THE HEATING AND VENTILATING MAGAZINE

VOL. VI., NO. 3 NEW YORK, MARCH, 1909 $1.00 a year, 10 cents a copy

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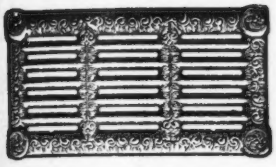

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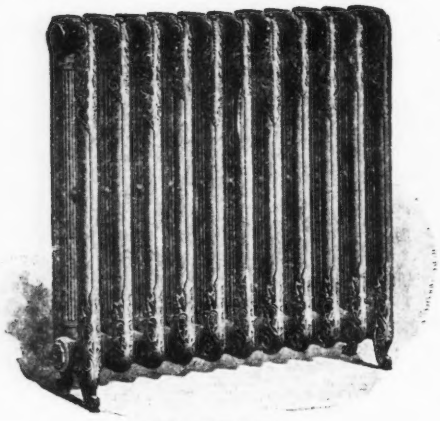

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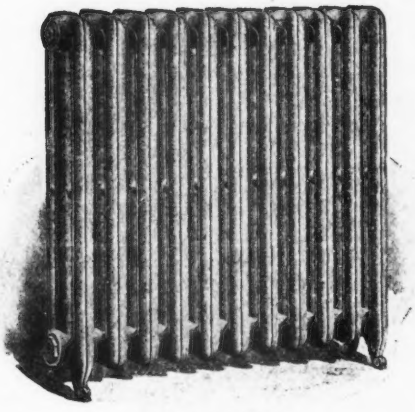

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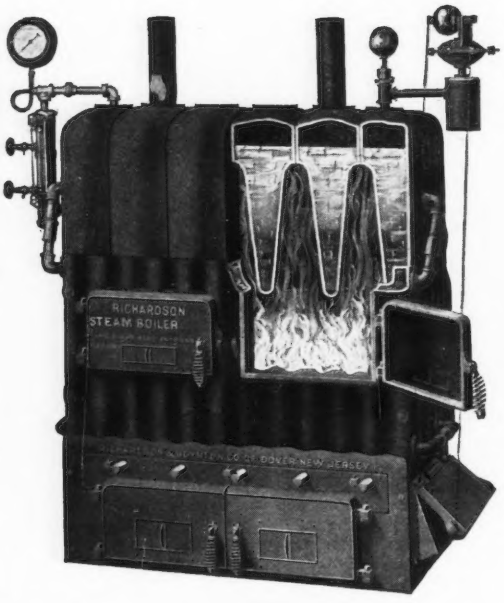

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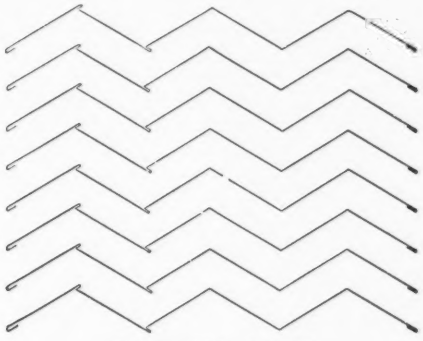

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The Heating and Ventilating Magazine

VOL. VI No. 3

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THE PEERLESS AIR WASHER

THE SPRAY HEAD

¶ The real cleaning of the atmosphere is attained by the original attraction of the moisture by the dust particles. The washing effect of the descending drops is only an incident in the process. The cloud particles permeate the space in which they are, the raindrops are relatively few and far between. Any apparatus which undertakes to remove the dust by passing the air through sheets of water or artificial rain is doomed to inefficiency because Nature's fundamental principle is overlooked. An atmosphere permeated with mist—water in its most minutely divided form—must be presented to the dust laden air. Each dust particle then becomes the center of attraction for the vapor particles, it is increased in size and weight, the air becomes supersaturated with water held in mechanical suspension.

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¶ This excess of moisture must be created in order to grasp and hold the solid matter. But once caught these particles may be easily separated from the air and conducted away with their burden of objectionable material. Hence the extractor or eliminator—a device designed to remove the mechanically suspended particles which represent the excess of moisture. In a word supersaturation and extraction are the two essential elements in the cleansing and purification of air. The air leaving the extractor should retain no visible moisture; only that which has been absorbed and with which it is saturated. The temperature within the spray chamber should be such that when the air is heated during its subsequent passage through the main heater or flue heaters the relative humidity will be sufficiently reduced.

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THE HEATING AND VENTILATING MAGAZINE

1123 BROADWAY NEW YORK MARCH, 1909

A Rational Formula Covering the Performance of Indirect Heating Surface

Being Part of a Discussion Before the American Society of Heating and Ventilating Engineers

BY PERRY WEST, M. E.

The formula (Formula 1) in its general form may be written: $R=K\frac{DZ^mN}{\sqrt[n]{V}}$ and for all practical purposes, within the scope and present accuracy on the subject it may be written $R=.085\frac{DZN}{\sqrt[3]{V}}$.

Here R = rise in temperature of the air.

K = a constant depending on the kind of heating surface.

D = an average of the summation of temperature differences, between the air and the steam.

$$=\frac{T_1-T_0}{\text{Log}_e\frac{T_S-T_0}{T_S-T_1}}$$

Z = the number of square feet of heating surface per square foot of clear area per unit depth of heater.

m = a power applicable to (Z) and depending on the type of heating surface.

N = number of units in depth of heater.

V = velocity of the air at 70° F. in feet per minute through the clear area.

n = a root applicable to V and depending on experiment.

The fundamental theory from which this formula was developed is the basing of the temperature rise of the air on the number of square feet of heating surface per square foot of clear area through the radiator, rather than upon the number of units in its depth. Another departure from common practice is in the introduction of the average of the summation of differences between the temperatures of the air and the steam as a factor, in place of either temperature difference between the entering air and steam, or the average air and steam.

The reasons for these departures seem reasonable enough and are as follows: The square feet of clear area through the heater when multiplied by the velocity gives the volume of air to be heated. Since the number of heat units required for this is practically in direct proportion to the volume, it is likewise in direct proportion to the clear area. Therefore the temperature rise will be in direct proportion to the number of square feet of surface per square foot of clear area. Again, it is a well established fact that the B. T. U., transmitted per square foot of surface per hour is practically directly proportional to the difference in temperature between the steam and air. This applies to all portions of the heating surface, hence

the average rate of transmission will be proportional to the average difference.

Also, since the temperature rise is proportional to this rate of transmission, it is in turn proportional to the average difference. The value for this average difference, D, as given above, may easily be deduced by the aid of calculus and has been used for years in the calculations for feed water heaters, which calculations are analogous to those for hot blast heaters.

It is believed that by the introduction of the above factors in the formation of the formulae given we get the basis, at least, for a rational formula and it only remains to assign the proper values to K, m and n from experimental data.

Formula No. 1 has been plotted out and is shown on Plate No. 1.

On Plate No. 2 is shown a comparison between this formula and some contemporary data of a similar nature.

All curves on this plate are shown for 1,000 ft. velocity, 0° F. entering air and 5 lbs. steam pressure.

This sheet is self explanatory and shows a very close relationship between the rational formula curve and some of the others under the particular conditions selected.

These conditions represent approximately average heating conditions and we would naturally expect all data of any value to check up more or less closely at these points.

Notwithstanding this, it is noted that some of the curves show widely differing values even under these average conditions.

Attention is drawn to the close agreement between Curves I, II and III, and to the fact that Curves II and III represent most recently developed and perhaps most accurate formulae and data in use to-day.

Curve II is plotted from the formula presented by one of your members at the last annual meeting, and will be recognized at once as

$$R = \frac{1}{\sqrt[3]{V}} (T_s - T_a) \frac{1}{\sqrt[8]{\quad} - .24}.$$

Curve III is plotted from the data recently given out by a reputable concern covering the performance of cast iron heating surface and checks up very closely with data of the same nature presented by your committee at this meeting.

These two curves, which agree with the rational curve so closely under the conditions named, have their limitations, however, and I think a word might be said concerning this and the shortcomings of some of the other data cited so as to present more clearly, if needs be, the necessity of some more accurate basis of figuring.

In Formula II, if we keep all conditions constant except the velocity, V, we get results varying directly as $\frac{1}{\sqrt[3]{V}}$. This would represent actual conditions only on the assumption that the average temperature difference between the air and the steam remained constant. This cannot occur, however, owing to the changes in R due to changes in velocity. The formula is an empirical one, therefore, which, while apparently correct for average conditions, will show undue variations for all others.

Curves III, IV and V are made up from manufacturer's tables and are rather outside of this discussion except for comparison. Some of this data evidently needs revision badly.

Formula VI and VII embody the same fault as Formula II, although in Formula VII this is somewhat ameliorated by using the average temperature of the air in place of the entering temperature of the same. Plate No. 3 will show the error to be expected from this. Besides this the two last formulae embody another more serious defect as follows: K, is a constant depending on the kind and type of radiation, on the number of units in the depth of the heater and on the velocity of the air. It has a number of values, therefore, each corresponding to a certain combination of the above mentioned variable factors. For any purposes, except approximations of the very crudest na-

ture, it is not only necessary to have a series of values for K corresponding to each kind and type of radiator, but that each series of such values shall cover all possible combinations between the number of units in depth of heater and the various air velocities.

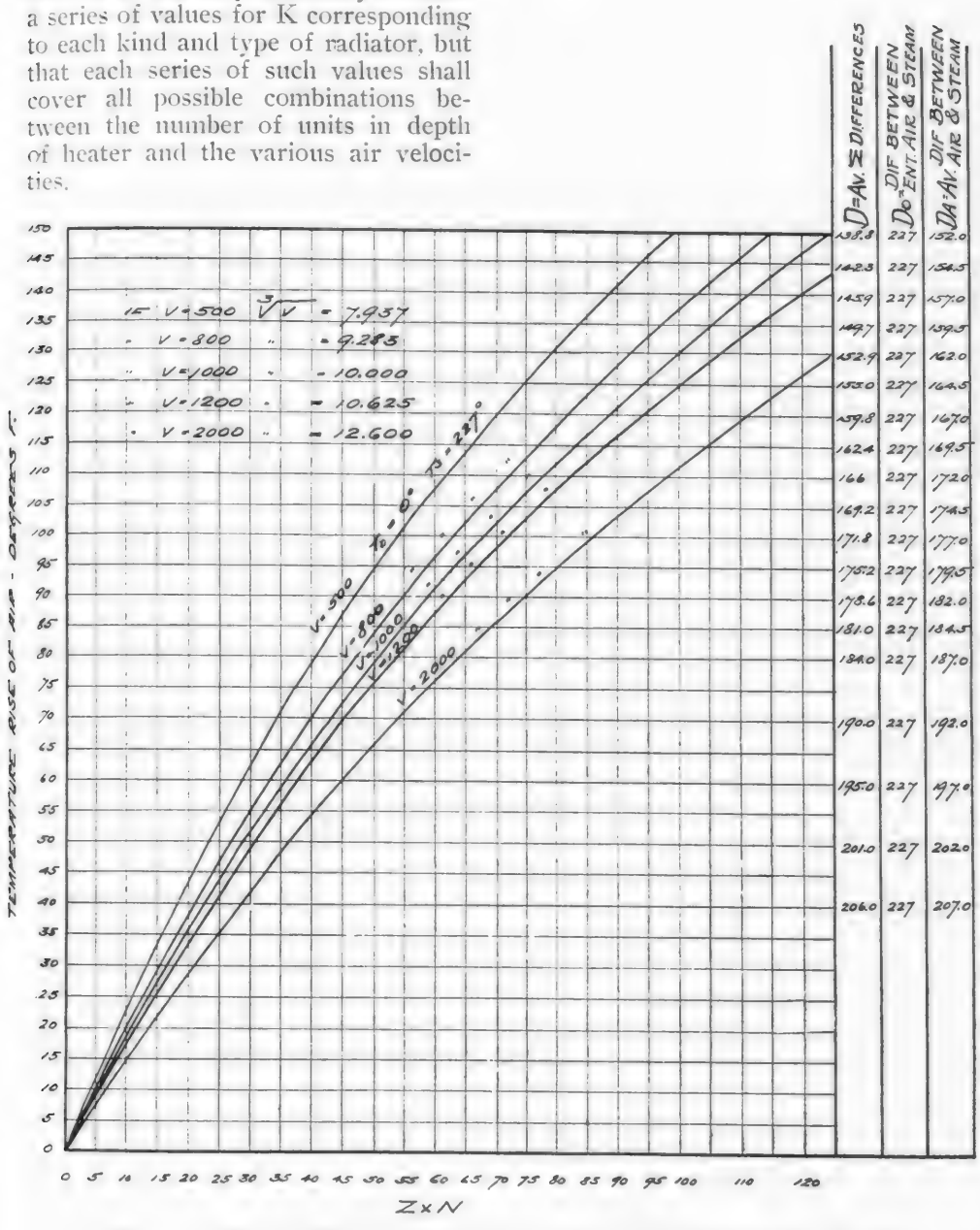


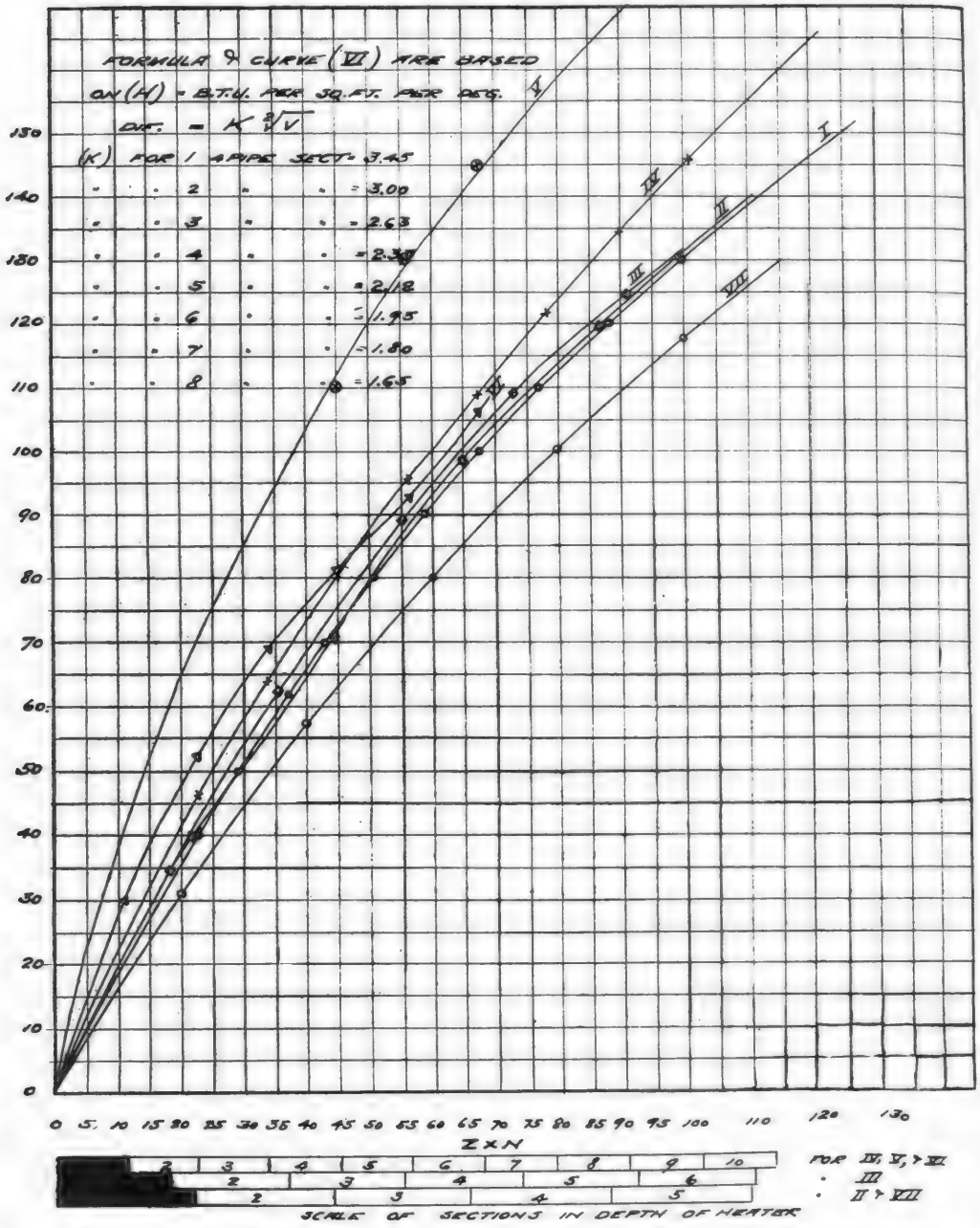
PLATE 1. CURVES SHOWING VALUES FOR R (RISE IN TEMPERATURE OF AIR.)

$$R = K \frac{D Z N}{\sqrt[3]{V}}$$

V = Velocity of Air in Feet Per Minute.
T_s = Temperature of Steam.
T_0 = Temperature of Entering Air.
T_1 = Temperature of Leaving Air.

D = Average of Σ Differences = $\dfrac{T_1 - T_0}{Log E \dfrac{T_s - T_0}{T_s - T_1}}$

Z = Square Feet of Heating Surface Per Square Foot of Clear Area Per Unit Depth of Heater.
N = Number of Units in Depth of Heater.
K = Constant = 0.085 for Above Curves.


FORMULA & CURVE (VI) ARE BASED
ON (K) = B.T.U. PER SQ. FT. PER DEG.
DIF. = K ∛V
(K) FOR 1 PIPE SECT. = 3.45
" " 2 " " = 3.00
" " 3 " " = 2.63
" " 4 " " = 2.37
" " 5 " " = 2.12
" " 6 " " = 1.95
" " 7 " " = 1.80
" " 8 " " = 1.65
0 5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100 110 120 130
Z X N
FOR IV, V, & XI
" III
" II & VIII
SCALE OF SECTIONS IN DEPTH OF HEATER

LARGE CURVES SHOWING COMPARISON BETWEEN DIFFERENT FAN BLAST DATA.

$V = 1{,}000$, $T_s = 227°$, $T_0 = 0°$.

Curve I = Rational Formula. $R = 0.085 \dfrac{D\,Z\,N}{\sqrt[3]{V}}$

Curve II = Rational Formula for Pipe Coils with 35% Clear Area. $R = \dfrac{1}{\sqrt[3]{V}} \left(T_s - T_0\right) \dfrac{}{S/N} \cdot 0.24$

Curve III = Manufacturers' Data on Cast-Iron Indirects (45% Clear Area).

Curve IV = Manufacturers' Data on 1-in. Pipe Coils (50% Clear Area).

Curve V = Manufacturers' Data on 1-in. Pipe Coils (50% Clear Area).

Curve VI = Formula $R = K_1 \dfrac{T_s - T_0}{\sqrt[3]{V}}$ (50% Clear Area Assumed).

Curve VII = Formula $R = K_2 \dfrac{T_s - \left(\dfrac{T_1 + T_2}{2}\right)}{\sqrt[3]{V}}$ (35% Clear Area Assumed).

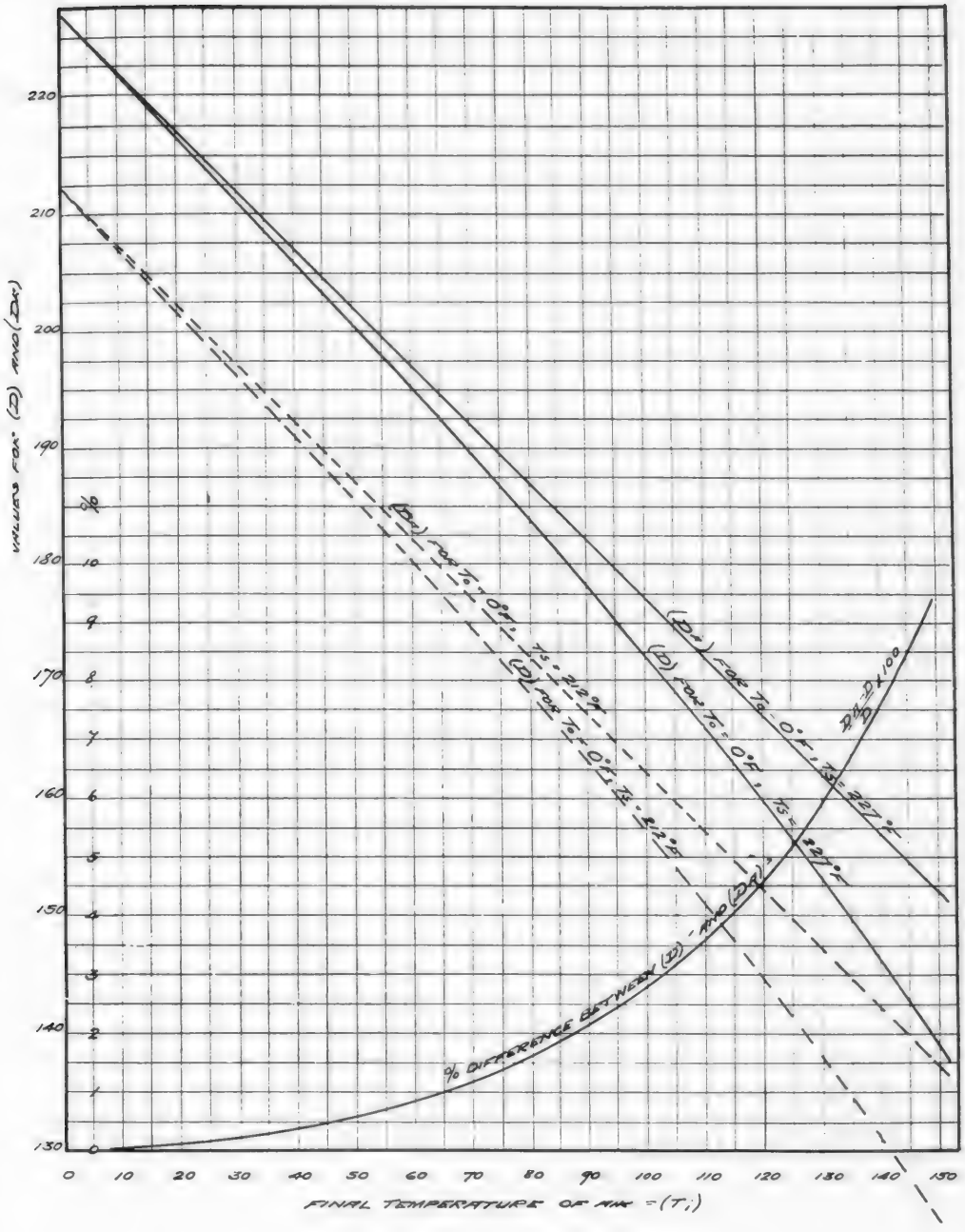

VALUES FOR (D) AND (Da)
FINAL TEMPERATURE OF AIR = (T1)
% DIFFERENCE BETWEEN (D) AND (Da)
(Da) FOR T0 = 0°F, T1 = 212°F
(D) FOR T0 = 0°F, T1 = 212°F
(Da) FOR T0 = 0°F, T1 = 227°F
(D) FOR T0 = 0°F, T1 = 227°F
% = Da/D x 100

PLATE 3. CURVES SHOWING VALUES FOR D AND D_a, ALSO THEIR PERCENTAGE OF DEVIATION.

D = Average of Summation of Actual Temp. Differences Between Air and Steam = $\dfrac{T_1 - T_0}{Log\text{E}\left(\dfrac{T_s - T_0}{T_s - T_1}\right)}$

D_a = Difference Between Average Temperature of Air and Temperature of Steam.
T_0 = Temperature of Entering Air.
T_1 = Temperature of Leaving Air.
T_s = Temperature of Steam.

It is a common occurrence to see this formula given with a table of values for K, depending solely on the number of units in depth of heater. In many such instances the kind of type of radiator is not referred to. When this last factor is omitted, it is, of course, folly for one to use the formula. This can be best appreciated by reference to any reputable heater tables, where it will be seen that almost twice the number of standard 1-in. pipe coil sections are required for the same temperature rise, with 50% clear area coils as with 35% clear area coils. As to the variations in K, due to changes in V, very little or no stress is placed upon this, as far as I am able to learn from the public records of the formula. As a matter of fact, I have nowhere been able to find the factor mentioned.

I do not know that I care to say more, except to ask that the worthy members present kindly apply any experimental data they may have to a closer determination of the undetermined factors and see if we cannot in some way arrive at a definite basis of figuring this performance of indirect heating surface.

The Effect of Wind on Heating and Ventilation

By H. W. Whitten

Supplementing two previous articles on the subject of heat losses around windows (published in The Heating and Ventilating Magazine for November, 1907, and October, 1908), Mr. Whitten discussed additional phases of the subject in a paper read at the annual meeting of The American Society of Heating and Ventilating Engineers in New York, January 19-21, 1909, in which he said:

For some years the writer has been a close observer of the action of wind on modern types of buildings as affecting heating and ventilating. The problem confronting the engineer in modern heating is to provide sufficient heat and air change in any weather condition liable to occur. It is clear that, if there are conditions in the building that are liable to be greatly affected by wind variation, he must estimate their effect on the interior of the building.

The most vulnerable places for the wind to exert its varying influence are the windows. These are usually made of the sliding type, and, in order to be readily movable in all seasons, must have at least 1/16 in. clearance all round. This clearance is usually greater. The inward or outward movement of air through these crevices is practically unimpeded and is nearly proportionate to outside conditions.

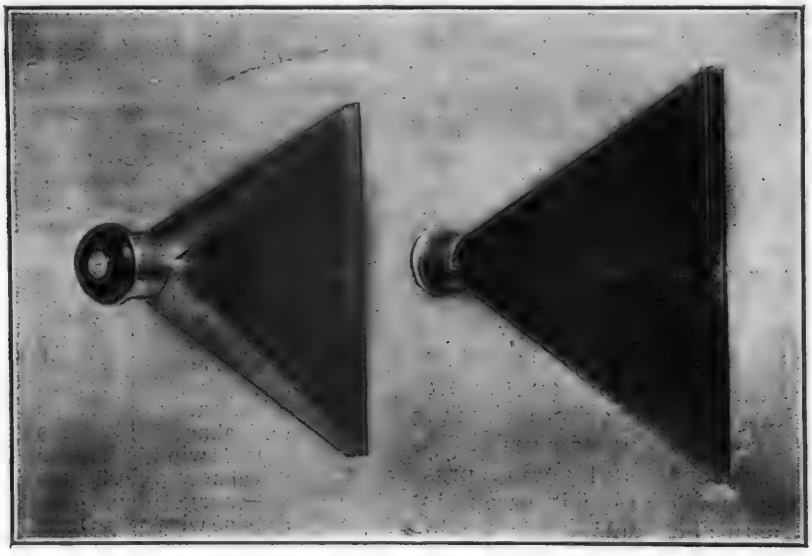
Tests made with a fan forcing air against a sash show that a pressure equal to a wind movement of fifteen miles per hour will force about 185 cu. ft. of air through one lineal foot of sash crevice 1/16 in. in width in one hour. Tests taken on sash in the open show that an average of 146 cu. ft. per lineal foot per hour comes through the same space when an actual wind of 15 miles per hour prevails.

The effect of this air leakage was tested on a steam radiator, but because of the variation of interior conditions of the radiator, due to the fact that the pressure varied at times, that the steam was sometimes wet and at other times dry, and that more or less air mixed with the circulation; an electric radiator was substituted. The heat from this radiator was fairly constant and could be readily checked. The action of air leakage on this radiator was readily apparent. A testing house was constructed, in which the walls were practically proof against heat loss. A window was placed in one side, which could easily be replaced by others of different clearance or with ones provided with wind-stopping devices. This house could be rotated so as to face prevailing winds.

These wind velocities and outside temperatures were recorded, also the time required to raise the interior tem-

perature by means of the electric radiator a certain number of degrees. The current being shut off, time observed during which the temperature fell to starting point. This operation was repeated until the walls practically ceased to absorb heat. The sash was then changed for others of different clearance, which were subjected to the same test. Sash with insulating devices were also tested.

These tests conclusively demonstrated that the average ratio of heat loss by leakage as the wind increased to decrease the rate of progression of loss and leakage, presumably because the pressure of the wind held the sash more tightly against the stops while its effect on the glass itself continued. It was evident from the investigations thus far pursued that, if windows were properly insulated, the radiation calculated against this heat loss from leakage could be practically eliminated, and the writer has followed this practice successfully on the basis of the prevention of inleakage. In further looking into this subject, however, an-



DEVICE TO MEASURE AIR LEAKAGE THROUGH ANY SECTION OF WINDOW JOINT

was, in the case of a sash with 1/16 in. clearance, 25% per mile, or for 4 miles increase in wind velocity the leakage heat loss increased 100%. In the case of the insulated sash the leakage heat loss increased at the average rate of 5% per mile of wind movement.

The actual comparative leakage heat loss between a sash with 1/16-in. clearance and the insulated sash, with a difference of 44° of temperature during a 15-mile wind, was as 5 to 1. The comparative cubic leakage being as 9 to 1.

The average wind movement in localities where heating is important is about 13 miles per hour during the heating season.

Increasing velocity of wind tended other and still more important phase of the effect of wind has appeared.

OUTWARD LEAKAGE OF AIR ON LEEWARD SIDES OF BUILDINGS

In order to test quickly the wind leakage in various buildings, particularly those where a dispute as to the efficacy of the apparatus occurred, an instrument was designed for the purpose of measuring leakage through any section of window joint, taking in about 2 lineal feet. The accompanying cut shows its construction. A 4-in. anemometer placed in the nozzle showed the air movement. Very many tests have been taken with this instrument, and have shown uniformly that the outflow from window crevices on the leeward sides of buildings exceeds

the inflow from the windward sides. Especially is this true in the case of buildings heated by the indirect method with forced circulation.

The expanded warmed air in the rooms on the windward sides of the building exerts a repelling force against inleakage, and is sufficient to overcome that caused by moderate winds of three to four miles per hour. Above this velocity inleakage occurs, increasing with the outer wind movement. This inleakage is met by the warmed air inflow, mainly in the upper portion of the room, and mixes with it to some degree. The greater portion falls by natural gravity to the floor and causes cold currents rising in spurts wherever it meets a warm human body, but the bulk of it is added to the vent flow. It has in the meantime upset the *calculated* circulation from the inlet to the vent.

In cases of this kind the thermometer will frequently register 70° on the wall and 58° on the floor. Direct radiation placed under the windows is a frequent makeshift to remedy this situation. The conditions on the leeward sides are worse from either a heating or ventilating standpoint.

Not being opposed by wind pressure, the expanded air will escape by its most natural channel, *i. e.*, the window crevice. It is usually introduced near the upper part of the room, and is *expected* to escape by means of the vent near the floor. Experiments have shown that the addition to the flow of the vent on the windward side from inleakage is much less than the subtraction from the flow of the vent on the leeward side caused by outleakage. In these cases the rooms were partitioned and a hall intervened.

The action of the wind blowing around and over the building is the undoubted cause of this acceleration of the outleakage. As it sweeps by, an area of low pressure is formed in the lee of the shelteerd side. This area exerts a siphonic action on every crevice in the sheltered side of the building, and is a continuous air pump on the interior supply of heat units. As most of these crevices are above the normal breathing level, the loss from them is absolute.

In a school building, recently examined by the writer in Pittsburg, it was found that the aggregate free opening around 150 windows averaging 4 ft. 6 in. in width by 7 ft. in height was approximately 46 sq. ft., or about 44 sq. in. per window. The required 30 cu. ft. of air per minute, even when supplied at a temperature of 120°, failed to warm this building during moderate winter weather.

TEST ON TOWER OF PUBLIC BUILDING, PHILADELPHIA

The result of a series of tests taken recently in the tower of the Public Building, Philadelphia, are as follows:

Average outside temperature, 38°.
Inside temperature, 70°.
Instrument shown in cut used with an anemometer 4 in. in diameter.
Average sash clearance 1/16 in.
Lineal inches of crevice covered by the instrument, 24.

Area of crevice		1.5 in.	
" " anemometer		12 in.	
Rate of outside wind	1,020	ft.	per min.
" " inleakage	125	"	"
" " outleakage	146	"	"
" " outside wind	975	"	"
" " inleakage	90	"	"
" " outleakage	109	"	"
" " outside wind	1,240	"	"
" " inleakage	119	"	"
" " outleakage	141	"	"
" " outside wind	1,420	"	"
" " inleakage	140	"	"
" " outleakage	158	"	"
" " outside wind	1,350	"	"
" " inleakage	132	"	"
" " outleakage	149	"	"

A comparison of these figures shows that the average outleakage is from 15% to 20% greater than the inleakage.

I have found that the average rate of inleakage in many buildings tested was about 10% less than that shown in the testing-house experiments and that the rate of outleakage was about 10% greater.

This is probably due to the fact that the cubic contents of the testing house (about 78 cu. ft), as compared with the lineal feet of window crevice in it (14 ft.) was less than average room conditions. A room with 16 ft. exposed wall, 12 ft. deep, with two windows 6 ft. high by 3 ft. wide, would be nearly typical. Such a room would

contain 1,920 cu. ft. and have 42 lin. ft. of window crevice. In the testing house the proportion of window crevice to cubic contents is 1 to 5.5. In the typical room mentioned it is approximately 1 to 46.

The effect of this greater volume of expanded air would be to retard the inleakage. Contrariwise, it would tend to accelerate the outleakage, which is also influenced by the suction from the area of low pressure existing on the sheltered sides of the building.

The writer had, previous to these tests, used, with good results, the formula published by Prof. John R. Allen. This formula is as follows:

$$\left(\frac{W}{4} + G\right) d\, n$$

W = square feet exposed wall.
G = square feet glass surface.
d = difference between inside and outside temperature.
n = leakage and wall filtration.

In this formula $n = 1.5$ to 2, according to conditions of wall construction, fitting of windows, etc.

With windows properly insulated, this formula may be modified as follows:

$$\left(\frac{W}{4} d\, n\right) + \left(G\, d\, n_1\right) \text{ in which } n_1 = 1.1.$$

This formula, while to some degree empirical, is more logical than any now in use which contain a variable factor for leakage. With window insulation the interior condition becomes stable and the maximum loss in any wind known. Artificial circulation is not interfered with or upset, and the control of inward or outward currents is established.

Evolution in Heating

V.

By J. M. W. Kitchen

In the last number I showed a special form of heater which, provides for using as much flue heating surface as may be desired. There was one point in this connection to which attention was not drawn. In most of the boilers now on the market there is relatively a very large amount of grate surface provided in proportion to the heating surface area. A very serious disadvantage of this disproportion is that, to secure heating capacity in an installation, too much grate capacity has to be provided, and this necessitates the production of too much heat during the many weeks of very mild weather in which a stand-by rate of combustion must be maintained to provide a small amount of heat during that time, and to be able to immediately have heat when a sudden lowering of the outside temperature occurs.

A certain amount of combustion must be kept up in order to keep the fuel mass in any furnace lighted, and if the grate surface is too large too much heat is necessarily generated in the apparatus during the mild period referred to.

Health, comfort and economy is induced by having a smaller grate, and providing more heating surface to pick up the heat generated by a more rapid rate of combustion with a smaller grate; it being understood that no more heating surface should be provided than is necessary for all ordinary weather. For the half a dozen blizzard days of the winter, in which a very high rate of combustion is required to meet the necessities of the case, it pays better to let some of the low degrees of heat go to waste than to provide sufficient flue heating surface to economize that heat during those half dozen days of extremely low temperature.

The student who undertakes the study of fuel economy in heating must, very surely, in the course of an evolutionary progress, be brought

face to face with an inquiry as to the routes through which heat wastes occur, and as to the possibilities of using regeneratively the thermic units that have once carried out the purpose of their generation.

As to the nature of the phenomenon of heat, we, of course, know very little. We refer to it as a form of force and in so dong express our ignorance. We know that it is a property of material bodies that is more or less concentrated, or more or less diffused. The forces of nature have concentrated this energy in the coal beds, and we set that energy free and diffuse it in the atmosphere in heating processes. We heat our buildings, and then throw most of this concentrated heat entirely away.

And yet, in theory, it is always practical to use this heat in cycle perpetually to a greater or less degree. The practicability of doing this in small installations is, of course, questionable; but, theoretically, is not necessarily impossible, especially in connection with mechanically induced ventilation. In more important installations the theory may be reduced to practice with decided economic benefit. As illustrating how this may be done, I present the design schematically shown in Fig. 13.*

THE TORRIDOOR REGENERATIVE HEATING SYSTEM

This system is designed for use in connection with schools, public buildings, hotels, churches and other public buildings. It introduces the principles of sectional radiation, heating through the floors, which are preferably formed of concrete or tile, by convection upwardly through the floor of some of the heat; of introducing air for respiration diffused through many apertures in the floors, or by drawing air downward through the floors; of heating one floor above another in the various stories, to a certain degree, with heat used in heating the spaces below the floors; of using the once breathed, effete, warmed air of the building, to heat new fresh air for respiration; heating that air primarily with the effected warmed air, secondarily with the water of condensation if the heating system is a steam system, or with partially cooled water if it is a water heating system, and of increasing the temperature of the incoming air finally with live steam or highly heated gases.

A study of the reference characters and of the parts referred to by them will give a sufficiently clear idea of the system.

A represents air passages for conveying fresh warmed air.
A^1 is a fan for forcing air throughout the building.
A^2 is an air meter.
A^3 is a damper for controlling over fuel-mass air for combustion.
A^4 is a damper for controlling the under grate air supply.
A^5 is a duct for warmed air for combustion.
B represents a water or steam boiler specially designed to burn bituminous coal and to carry out the principle of the counter-current travel of the gases and the water.
B^1 is a section of the boiler B.
B^2 represents steam conducting pipes.
B^3 is a feed water pipe.
B^4 is a steam dome.
C is a combustion chamber having an exceptionally high run for the gases.
D are foul air ducts.
D^1 are entrances to the ducts D.
E is an air heating economizer, introducing the principle of the counter-current transferrence of waste heat to the newly introduced air.
E^1 is a section of the economizer E, for economizing the heat of the respired air of the spaces occupied in the building.
E^2 is a section for economizing the waste heat of the water of condensation, and cooling the water to give it better recipient temperature as feed water.
E^3 is a section for economizing the waste heat of the gases of combustion.
F is an air flue for introducing cool air into the system.
F^1 is an air injecting device for cool air.
G is a catch basin which contains a non-odorous antiseptic solution to disinfect dirt falling through the register faces in the floors.
H is a return pipe for the water of condensation.
I are air heating tubes.
J is a suspended ceiling made of metallic movable pannelling, covering the piping suspended below each floor.
K is a suspension frame.
L is a register outlet.

*The design herewith shown is controlled by the Torridoor Power, Heat and Lighting Co. of America.

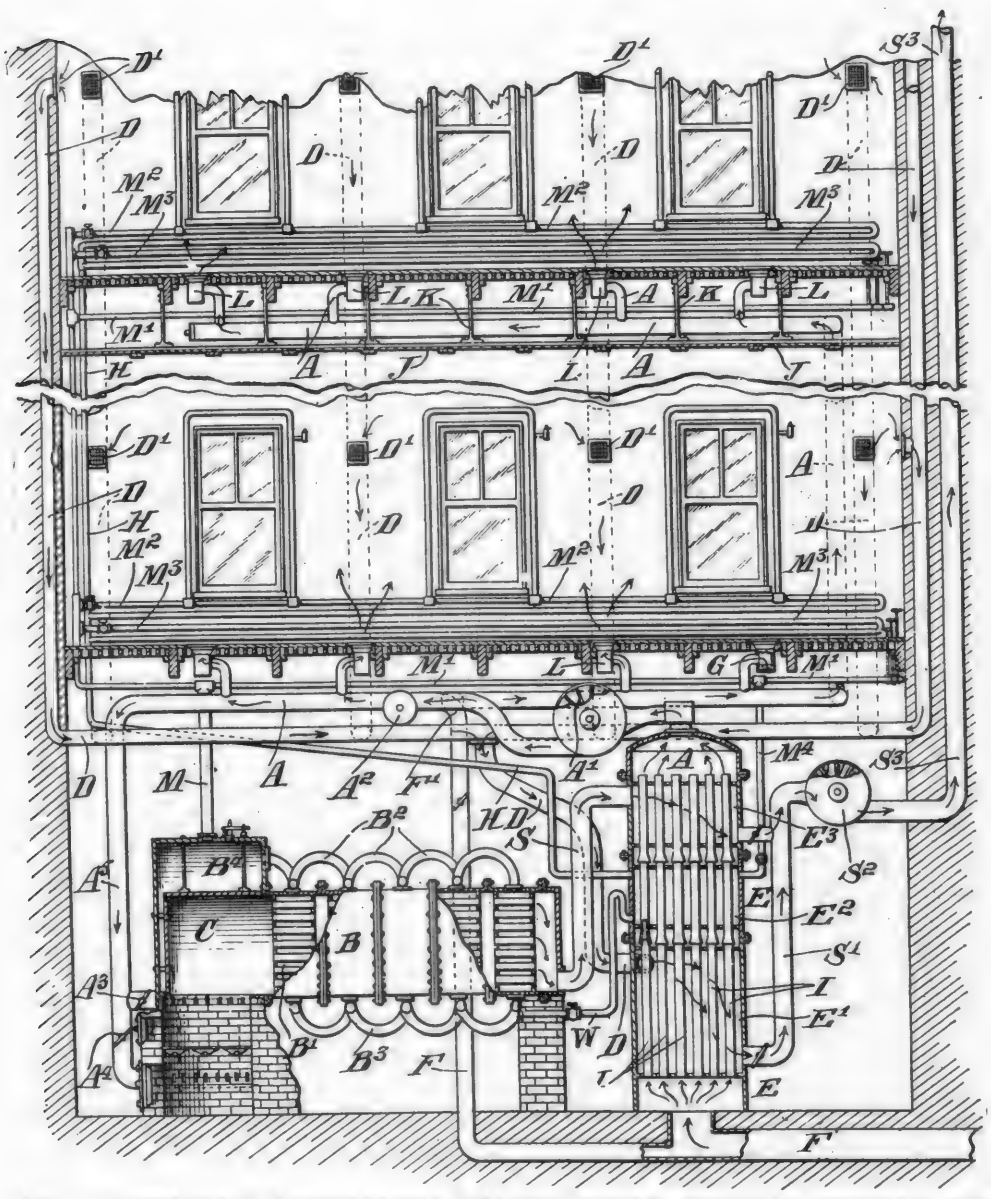


FIG. 13.—TORRIDOOR REGENERATIVE HEATING SYSTEM SPECIALLY DESIGNED FOR SCHOOL HOUSES

M is a steam main.
M^1 is a secondary steam conduit.
M^2 is a sectional steam conduit.
M^3 is another sectional steam conduit.
M^4 is a pipe for conveying live steam.
R is a register face.
S is a smoke conduit.
S^1 is an exit conduit.
S^2 is an induction fan.
S^3 is a waste gas conduit.
W is a feed water pipe for the boiler B.

In the steam or water distributing system, the piping is divided into sections, each section being controlled by a valve, so that one of more section can be brought into use at the same time, or excluded as radiating surface. Under the several floors each section of piping is placed so that there is substantially uniform distribution of the radiated heat under each floor; the total amount of heat radiated being governed by bringing one or more sections into action.

In ordinary practice it is desirable to have at least three sections of pip-

ing in the system. Those surfaces of the building which are exposed to special degrees of radiation are also supplied with direct steam radiating sections individually controlled by valves.

It will be observed that this system enables the principle to be carried out of having the feet warm and the head cool; that each individual has his own individual fresh supply of air to breathe, and that air once breathed is not breathed again in the building.

It is a debatable question as to whether it is better hygienically to introduce the air at a high level and draw it downward through the floor, or to have it forced upward through the floors; but, individually, I prefer the latter method. The special floor provided for in this case is intended for the use of movable desks and seats, and for the daily mopping of the floors with antiseptic solutions. In case the seats and desks are permanently fastened, I should prefer the ordinary mushroom arrangement of the air duct outlets. The arrows in the drawing indicate the travel of ingoing fresh air and outgoing effete air. Of course this travel could be reversed if desired.

The system here shown provides for the heating of a building with a very large reduction in fuel costs, a cost that is probably only one-quarter of the cost that is usual in heating school houses at the present time. The provision for a paneled ceiling below the piping hung from and below each floor makes this system practical when without the ceiling to hide the piping aesthetic reasons would make the use of the system objectionable.

The important matters of health and comfort are well looked after. It will be understood that the convective flooring is only moderately heated, and that the relative amount of heat radiated from the floors would, when needed, be increased to a suitable degree by the heat conveyed to the air for respiration, mechanically forced throughout the building.

The motor appliances in a system of this kind might be actuated by an electric current derived from outside, or by power generated in the system. In a later article, which I will present next month and in which I introduce the idea of using producer gas in heating systems, and burning a very considerable amount of that gas in an internal combustion engine for securing a perfect combustion, I will show how the kinetic energy developed by the engine would be utilized for running the ventilating fans.

(To be continued.)

Death of Edwin H. Roberts

Edwin H. Roberts, president of the Standard Steam Specialty Co., New York, manufacturer of the Utility line of feed water heaters and other heating supplies, died February 27th last of pneumonia and spinal meningitis. Mr. Roberts was on a visit to his mother in Auburn, Me., when he was taken ill with pneumonia. Although in a serious condition his illness was not considered critical until it was found to be complicated with spinal meningitis. He grew rapidly worse and died at Auburn within a week of his arrival. The body was taken to his former home in Somersworth, N. H., where the burial took place March 1st.

The pall bearers were C. N. Brizse, Norman Brizse, John L. Widder, P. Pinder, William Long and Herbert A. Joslin.

Mr. Roberts was 44 years old. He was one of the incorporators of the firm of Geo. I. Roberts & Bros., New York, jobbers of steam fitting supplies being a brother of Geo. I. Roberts, and at the time of his death was treasurer of the company.

Few men in the heating trade, especially in the East, were better known or more deeply respected than Mr. Roberts. A man of refined instincts and broad business principles, he contributed much to the maintenance of the best standards of the trade, while personally he was a man whose gentle bearing and natural generosity won for him a very large circle of friends. During recent years Mr. Roberts was a prominent figure at conventions of the National Association of Master Steam and Hot Water Fitters where he was a leader in the social features of the meetings, especially the golf events, as he was an enthusiastic golfer. He is survived by a widow, who is also well-known to convention goers as a prominent participant in the ladies' social events. She will have the deep sympathy in her sad bereavement of all who knew her husband.

Government Tests of Coal as Fuel for House Heating Boilers

1—Tests on Sectional Steam Boiler at St. Louis Plant
2—Tests on Sectional Cast-Iron and Round Boilers at Urbana, Ill.

The United States Geological Survey has lately completed a valuable service to the heating trade by making a series of tests of coal and briquets as fuel for house heating boilers. So exhaustive were the tests that it has been possible to prepare a table showing the relative value of different fuels for domestic purposes. From this table it is possible to arrive at the probable value of any fuel in any part of the country.

The figures in the table show that with a sectional boiler the effectiveness of different fuels depends on the number of thermal units they contain. This means that if all fuels cost the same, the cheapest would be the one containing the largest number of thermal units. One basis for ascertaining the relative value of different fuels lies in the cost of evaporating 1,000 lbs. of water. If, for example, at a certain place anthracite costs $8 a ton and Pocahontas bituminous coal $4, the cost of evaporating 1,000 lbs. of water with anthracite would be 52 cents and with the bituminous coal 23.68 cents. This shows that a saving of about $4 a ton for the same amount of work done would be effected by purchasing the bituminous coal.

The data obtained in the tests show that as the percentage of the rated capacity developed by the furnace increases the percentage of efficiency increases; also that high capacities were not obtained with the good coals, nor were the low capacities the result of burning poor coals.

In the following review of the tests the results on briquets will be omitted in view of the fact that a comparison of results on coal and briquets shows no advantage in the briquets over coal of a size larger than screened nut. Briquetting good bituminous coal would be justified, according to the report, only when slack is used for material. The briquets tested gave much smoke, which was due to the pitch used as a binder.

Both series of tests are reported by D. T. Randall. They were begun in October, 1906, in St. Louis, and in the spring 1908 in Urbana, at the University of Illinois engineering experiment station. The equipment at Urbana permitted more uniform conditions of pressure and capacity, making the results more valuable for comparison.

The St. Louis Tests

The boiler on which the tests were made at St. Louis is a sectional steam boiler containing seven separate sections; it is 66 in. long and stands on a foundation 47 x 63 in. The grate measures 36 x 54⅛ in. The boiler has three 5-in. steam outlets at the top of the sections, and is rated to take care of 3,150 sq. ft. of radiating surface. The elevation and plan of the St. Louis plant are shown in Figs. 1 and 2.

The boiler furnished steam to two buildings, supplying the necessary amount through two of the three outlets. The front outlet was not in use and was capped over. This boiler is made so that feed water may be supplied to every other section, entering at the base of the section on both sides, just above the grate level. There are six return inlets, but in this installation only the rear two were used. The piping was so arranged that during a test period all of the condensed steam was returned to the weigh tank, then allowed to discharge into the supply tank, from which it was forced into the boiler by a hand pump. The temperature of the water entering the boiler ranged from 100° to 150° F., but usually averaged 140°. During a test all the water entered through one of the rear inlets.

In a sectional boiler as usually installed the steam as it is taken from the sections is first drawn into a collecting drum, so that in case of a sudden demand for steam, water will not be carried into the heating pipes. There was not enough head room for the installation of such a drum at the

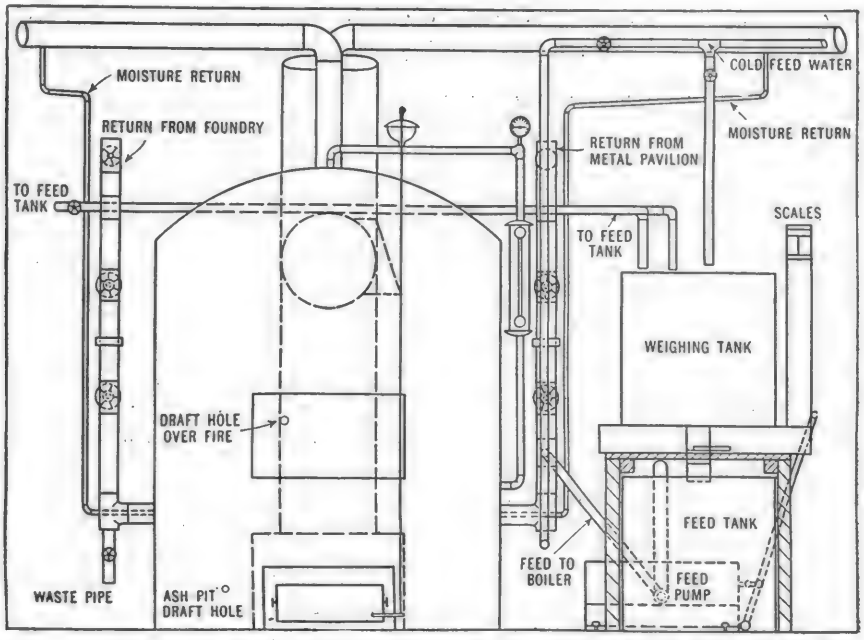


FIG. 1.—ELEVATION OF HOUSE HEATING BOILER PLANT AT ST. LOUIS, MO.

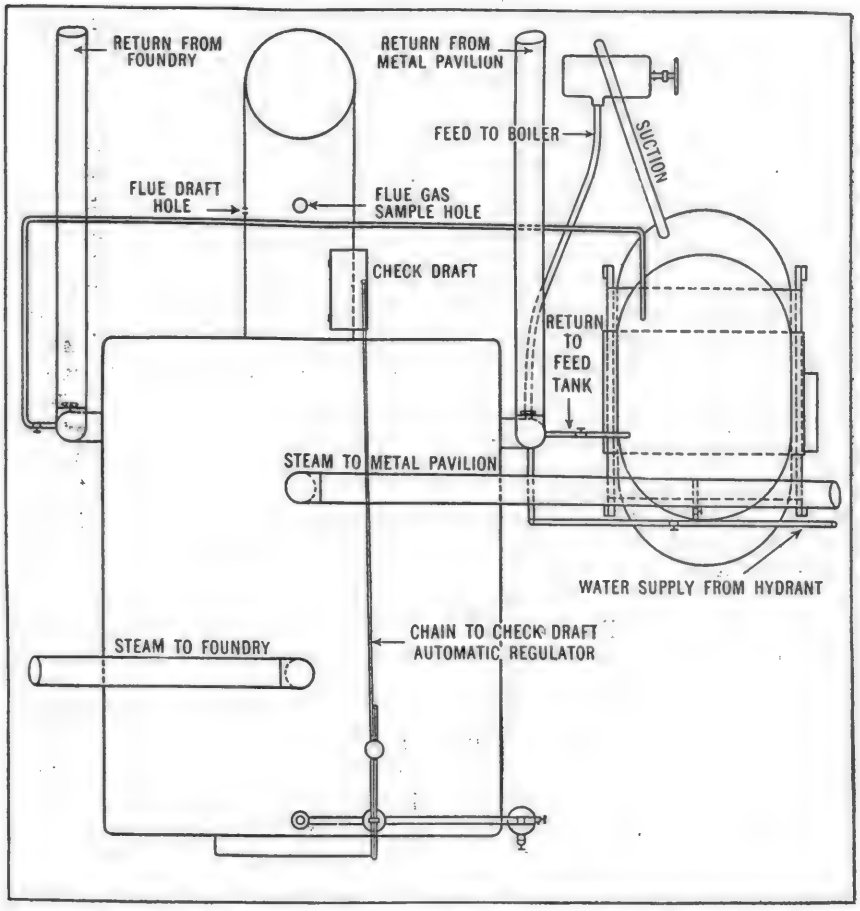


FIG. 2.—PLAN OF HOUSE HEATING BOILER PLANT AT ST. LOUIS, MO.

St. Louis plant, and the steam was drawn directly from two sections. Drainpipes connected with the bottom of the steam pipes, as shown in Fig. 1, carried moisture back to the boiler. No readings of moisture in the steam were taken, and therefore, in calculating water evaporated, the boiler was credited with slightly more work than it performed.

An automatic damper regulator was used to control the draft through the fire, so that a nearly constant steam pressure might be carried. This regulator was connected directly to the lower check and, by an arrangement of pulleys, to the check in the flue. All the exposed surface of the boiler except the front was lagged with about an inch of plastic asbestos covering.

In the first 30 tests the draft was supplied by a stack 32.4 ft. high above the grate. In these tests both natural and induced draft were used. In the last 28 tests the stack was raised to a height of 44.3 ft. above the grate level. This height always furnished the required draft.

The tests were made as nearly as possible under the actual running conditions of the average house heating boiler plant. Steam was supplied to two buildings for heating, and consequently the load varied with the weather conditions—the temperature of the outside air and the velocity of the wind. On only a few of the tests was the heating load so light that steam was turned into the atmosphere.

Most of the tests covered a period of about 8 hrs.; during this time the operator tried to maintain a steam pressure of about 3 lbs.

The alternate method, as prescribed by the American Society of Mechanical Engineers' code for making boiler trials, was used in starting and stopping the tests. The boiler was installed in so small a room that the standard method of starting and stopping was not practicable. Each test was started with a fire about 4 in. thick. This thickness was gradually increased to a maximum of 12 to 18 in., depending on the kind of fuel burned and on the judgment of the operator as to the best thickness for good combustion. To start a test with a 4-in. fuel bed required that the boiler should be fired with the same kind of coal as was used on the test for about 4 hrs. before beginning the test. In building up the fire and in burning it down for a close light firing was used. At other times during the tests the charges of coal were considerably greater. Coal was fired whenever the heat requirements demanded it, usually at a time when the fire had burned down and the steam pressure had dropped. The fresh coal was spread over the entire fuel bed. The fire seldom required any attention, and it was never poked unless a coal in burning caked badly. On the tests the fire was cleaned only just before starting and stopping, except in two or three tests when there was an unusually large accumulation of clinker upon the grate.

Readings were taken of draft, temperature, and steam pressure every 30 min. Smoke readings were taken as soon after coal was fired as other observations would permit, and at intervals thereafter until the volatile matter had distilled and no smoke was given off for the stack.

Owing to the many duties of the observer there was difficulty in procuring and analyzing representative samples of flue gas, but the results are considered of sufficient accuracy to indicate certain general relations between the air supply and the performance of the boiler. All gas samples analyzed showed some CO. This was to be expected, for the combustion was never complete on any of the tests, as could be seen by watching the top of the stack.

DEDUCTIONS FROM TEST DATA

Table 1 is a compilation of nine different items from the St. Louis tests (exclusive of tests on mixed coals and one on Arkansas No. 13) averaged on the percentage of volatile matter in the combustible as a base. These results emphasize the difficulty of burning high-volatile coal in the fire pot of the average house heating boiler, which has either inefficient or insufficient combustion space. The vol-

atile matter varied over a wide range. The proximate analysis of fuel as fired showed that the lowest percentage of volatile matter was 14, in a Virginia coal, and the highest 38, in an Indiana coal. The efficiency decreased regularly and decidedly as the volatile matter increased. The drop in efficiency was undoubtedly hastened by clinker formation, the percentage of clinker increasing with increase of sulphur values. Both the color and the analyses of the gases of combustion indicated lessening completeness of combustion as the percentage of volatile matter increased. The table shows that the volatile matter was neither an index of the CO_2 in the flue gas nor of the rated capacity developed, but this was to be expected, for each coal was made to carry the load required by weather conditions and was not burned at its highest possible rate of combustion.

Table 2 was derived by averaging the efficiency obtained in 52 tests on the percentage of rated capacity developed as a base. A study of table 2 shows that as the capacity increased the percentage of efficiency increased. The high capacities were not obtained with the good coals nor were the low capacities a result of burning poor coal. This relation of efficiency and capacity might be expected when it is remembered that to burn sufficient coal to develop high capacity the available draft must be used, giving more air for the combustion of the coal.

When the boiler is run at low capacities the lower check draft is closed a large part of the time and, although this is effective in reducing the rate of combustion, the coal that burns is consumed with a very small air supply, resulting in a loss of combustible gases and making both the boiler and furnace inefficient.

Table 3, based on 52 of the tests, shows the loss in efficiency when running with a diminished air supply. This table was obtained by averaging the percentage of CO in the flue gas and the efficiencies on the percentage of CO_2 in the flue gas as a base. Table 3 shows that as the CO_2 increased the CO values became higher, but not with regularity, probably owing to differences in the character of the fuels.

TABLE 1.—AVERAGE OF RESULTS FROM 50 TESTS, BASED ON PERCENTAGE OF VOLATILE MATTER IN THE COMBUSTIBLE

Number of tests averaged.	Volatile matter in the combustible.	Sulphur in dry coal.	Ash in dry coal.	Efficiency.	Clinker.	Black smoke.	Percentage of rated capacity developed.	CO_2 in dry flue gases.	CO in dry flue gases.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*	*Per cent.*	*Per ct.*		*Per cent.*	*Per cent.*
4	18.30	1.33	8.00	60.56	0	18.2	63.7	8.7	0.44
12	22.71	1.74	8.94	56.33	0	18.0	61.1	7.3	.50
7	34.70	1.23	11.27	54.11	2.9	22.1	52.8	7.4	.55
11	38.79	2.64	15.02	47.19	12.0	30.8	57.1	8.7	.62
16	44.46	4.35	14.57	47.19	30.4	32.9	61.3	9.1	.74

TABLE 2.—RELATION BETWEEN PERCENTAGE OF RATED CAPACITY DEVELOPED AND EFFICIENCY

	Number of tests averaged for each point.	Percentage of rated capacity developed.	Efficiency.
			Per cent.
Group I	1	44	43.5
Group II	3	48	48.5
Group III	12	53	47.8
Group IV	20	58	52.5
Group V	9	64	53.3
Group VI	5	69	53.7
Group VII	2	76	62.7

TABLE 3.—RELATION BETWEEN CO_2 AND CO IN FLUE GAS AND EFFICIENCY

	Number of tests averaged.	CO_2 in flue gas.	CO in flue gas.	Boiler efficiency.
		Per cent.	*Per cent.*	*Per cent.*
Group I	2	5.8	0.12	55.3
Group II	6	6.7	.36	51.6
Group III	17	7.4	.42	53.4
Group IV	9	8.4	.82	49.3
Group V	11	9.6	.54	48.2
Group VI	4	10.4	.94	59.4
Group VII	2	12.3	.75	45.8
Group VIII	1	13.3	.60	45.2

However, the low CO averages appear with the low CO_2 averages and vice versa. This is sufficient proof that the combustion becomes less complete with a reduction of the air supply, resulting in higher values of CO_2. It is noticeable that at all times there was a harmful amount of CO present. The table demonstrates that with only one exception the boiler efficiency decreased with increase of CO_2, so that when the value of 13.3% CO_2 had been reached the efficiency had dropped to 45.0, its lowest point. That high CO_2 in the flue gas is not necessarily an indication of high economy was pointed out in a previous report.

TABLE 4.—HEAT BALANCES, OR DISTRIBUTION OF THE HEATING VALUES OF THE COMBUSTIBLES FROM BOILER TESTS

Kind of boiler	House-heating.			Power plant.
Test No.	39.	31.	56.	56.
Designation of sample	Ill. No. 9 C.	Pa. No. 18.	Pa. No. 18 and Misc. No. 9.	W. Va. No. 11.
Total heat value of 1 pound of combustible, British thermal unit	14,428.	15,842.	14,947.	15,934.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Heat absorbed by boiler	40.39	56.51	56.48	71.21
Loss due to—				
Moisture in coal	.65	.43	.11	.46
Moisture formed by burning of hydrogen	4.53	3.43	2.72	3.24
Heat carried away in dry chimney gases	25.91	22.82	24.93	19.37
Incomplete combustion of carbon	5.75	2.51	2.35	.33
Radiation unaccounted for	22.77	14.30	13.41	5.39
	100.00	100.00	100.00	100.00

It is important to note that in the tests showing the lowest CO_2, the most air entered the furnace, the coal was more nearly burned, and the highest boiler efficiency was obtained. The table shows that the efficiency was higher with low CO values and accompanying excess of air than with reverse conditions. In all the tests there was not sufficient air used.

Table 4, gives three heat balances on the house heating boiler and one on a 210-HP. hand-fired Heine boiler, determined at the fuel-testing plant at St. Louis. Test 39 was taken because it was a low efficiency test with a high average flue-gas temperature; test 31 was a high-efficiency test with a low flue-gas temperature; and test 56 was a high-efficiency test, with a high flue-gas temperature. The test on the high presure boiler was added for comparison of the results obtained when burning coal in a small boiler and a reasonably large one. This test gave a higher efficiency than may be expected of the average commercial plant; it shows, however, what may be accomplished in the large plant as compared with the small one. The three loss items which differ the most are the loss up the stack, loss due to incomplete combustion of carbon, and radiation and unaccounted-for loss. These items must be higher on the house heating boiler owing to the higher flue-gas temperature and poorer combustion.

The high flue-gas temperature is due to inefficient heating surface, either poorly arranged or made inefficient by the soot and other products of incomplete combustion. Bulletins describing the work of the fuel-testing plant have called attention to the fact that the CO loss was indicative only of the true loss due to incomplete combustion and might not be nearly so large as the unknown loss. In the tests of Table 4, as the CO loss increases both the unaccounted-for loss and the loss up the stack increase; this corroborates former work of the Survey on high-pressure boilers and emphasizes the importance of con-

structing furnaces so that the fuel may be burned with a sufficient amount of air, properly distributed.

Another table was derived by averaging the percentage of clinker in refuse and of black smoke on the percentage of CO_2 in flue-gas as a base. Although it shows, with but one exception, that the CO_2 increases as the clinker in the refuse increases, the reader must not infer that the reduced air supply was entirely the result of clinker formation. It is true that when clinker forms on the grate there is more difficulty in drawing air through the fuel bed than when coal burns to a free ash; however, on these tests the lower check draft was open only from a quarter to half the time, so that during the major portion of each test the coal was burned with reduced air supply. It is interesting to note from this table and from Table 1, that the smoke increased in density as the percentage of clinker in the refuse increased.

The Urbana Tests

The two house-heating boilers in the experiment station at Urbana are designated in this report as boiler A and boiler B. Both boilers are standard types, supplied for use with either water or steam heating systems, except that in boilers of the type of boiler A a double section is usually substituted for the single intermediate section when used for water heating.

The general arrangement of the plant is shown in Fig. 3. The boilers are set independently, and each is provided with similar load regulators, return feed systems, and stacks. The flow is so arranged that the steam may be discharged to the atmosphere through an exhaust head above the roof of the building or into heating coils at the rear of the boilers. The coils contain 1,000 sq. ft. of radiating surface, and are arranged in six sections, any number of which may be cut out, and the amount of radiation changed in proportion. They constitute, with the overhead discharge pipes, the regular heating system of the experiment-station laboratory.

Boiler A is built up of four cast-iron sections—the base and grate section, the fire pot, the intermediate circulation section, and the dome. The water space surrounds the firepot, and is continued into the intermediate and dome sections through three nipples.

Boiler B represents a common type of sectional construction in which the

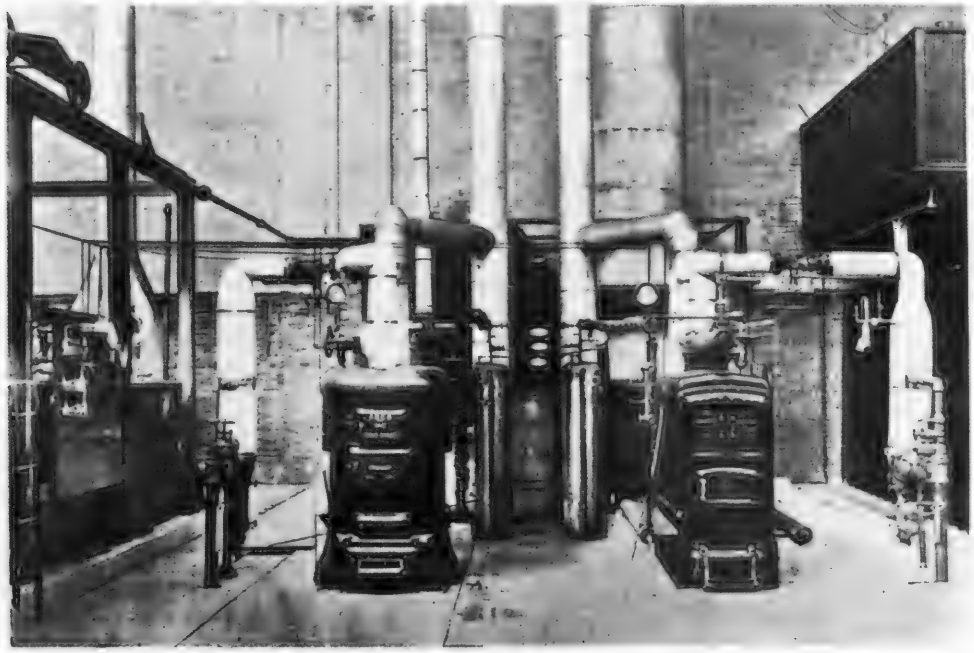


FIG. 3.—HOUSE HEATING BOILER PLANT AT URBANA, ILL.

base or grate portion and the water-heating portion are built up of interchangeable cast-iron sections, the water legs or sections being connected by means of external circulation drums or headers. The principal dimensions and proportions of both boilers are given in the accompanying table. (Table 5.)

TABLE 5.—DETAILS OF THE BOILERS

	A.	B.
Rated capacity, radiating surface, square feet	800	1,075
Height over all, feet	5½	5¾
Floor space, square feet	9	25
Size of fire door, inches	8½ x 15	9 x 15
Height of fire door above grate, inches	14	10
Fuel capacity to center of fire door, pounds	290	370
Kind of grate	Plain rocking.	Rocker.
Width of grate, inches	28	18
Length of grate, inches	28	48
Area of grate surface, square feet	4.28	6
Area of air space, square feet	2.15	3
Ratio of air space to grate surface, per cent	50	50
Mean height of furnace, inches	22.5	22
Height of chimney above grate, feet	39	39
Sectional area of chimney, square feet	1.07	1.07
Area of flue connecting to chimney, square feet	0.55	0.55
Length of flue connecting to chimney, feet	14	12½
Least flue area in boiler, square feet	0.67	0.495
Ratio of smallest flue to grate surface, per cent	15.5	8.23
Kind of draft	Natural.	Natural.
Direct water-heating surface, square feet	18.8	21.89
Indirect water-heating surface, square feet	20.7	53.98
Superheating surface, square feet	4.2	None.
Total heating surface, square feet	43.7	75.87
Ratio of direct-heating surface to total, per cent	43	28.9
Ratio of total heating surface to grate surface	10.2 to 1	12.6 to 1
Total water and steam space, cubic feet	7.38	11.16
Steam space, cubic feet	3.07	3.28
Water space, cubic feet	4.31	7.88
Area of external boiler surface in contact with water or steam, square feet	37.88	103.27
Smoke outlet above grate, feet	4.17	3
Diameter of flue, inches	14	14

The equipment of these small boilers for experiments necessitated an especially devised method of supplying feed water and of controlling the load demand.

RETURNING CONDENSATION TO BOILERS BY COMPRESSED AIR

As the condensation return had to be intercepted for the purpose of weighing or measuring the feed water, the usual method of returning the condensed steam by gravity could not be used and other means had to be provided for forcing the water back into the boiler. A number of methods might have been used provided care was taken to insure steady feeding. The decision to adopt the system by which the feed water is forced from the measuring tanks into the boiler with compressed air was influenced to some degree by the temporary necessity of procuring condensation water from the heating coils. As all of the apparatus is located on the same level, this system has proved convenient and satisfactory.

The tanks used are ordinary galvanized-iron range tanks. One supply tank, with a capacity of 54 gals., is connected directly in the return from the heating coils. The inlet pipe is at the bottom and the outlet at the top, so that the tank is always filled with hot water. The measuring tanks, or feed tanks, one for each boiler, have a capacity of 35 gals. each and are fitted with gauge glasses and scales graduated to read pounds direct, correction being made for varying temperatures. An overflow pipe, with valve, placed at the top of the scale, allows filling the tank with a definite charge of water.

These tanks, as stated above, are

connected with the compressed-air system and during feeding are under pressure sufficient to force the water into the boilers, a ¾-in. needle valve near the boiler allowing close regulation of the feed. The heating system is under 3 to 5 lbs. pressure, so that the measuring tanks are rapidly filled by simply opening a valve leading to the supply tank. The air pressure and

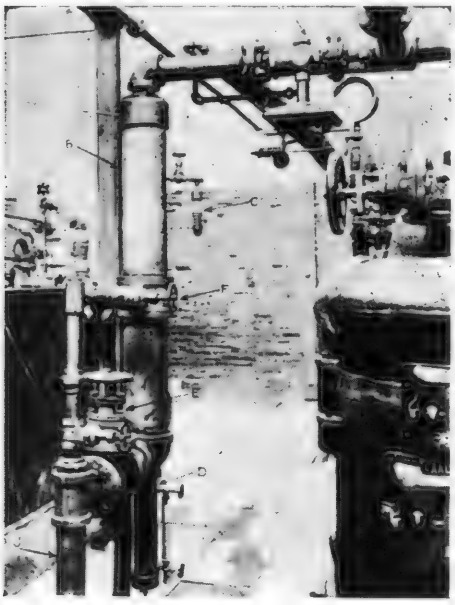


FIG. 4.—LOAD REGULATOR AT HOUSE HEATING BOILER PLANT AT URBANA, ILL.

boiler are cut off and the overflow is opened during charging.

Facility of controlling the rate of combustion is important when comparative studies of fuels are to be made. However, as these small heating boilers are designed to regulate themselves, under control of their automatic damper regulators, the rate of combustion may be kept fairly even by keeping the rate of evaporation constant. This has been accomplished at Urbana by the use of a pressure regulator, by which a steady flow of steam is discharged from a constant-pressure receiver through a suitable orifice into the atmosphere, provision being made for varying the load to suit the specific demand of the test. The receivers perform the duties of separators and are thus used to replace the usual steam calorimeter; for this reason the receivers and pipes are heavily lagged with 1 in. of hair felt laid next to the iron and above this 1 in. of magnesia pipe covering.

DESCRIPTION OF LOAD REGULATOR

Fig. 4 shows one of the load regulators with its covering removed. The steam from the boiler passes through the pressure regulator A into a 3-in. pipe which extends through the top of the receiver and nearly to the reducing tee at the bottom. Here the direction of the steam is changed and the entrained moisture separated, the dry steam passing up and out through the valve F to the exhaust main J, through an orifice plate in the 2-in. union G, the pipe J being open to the atmosphere. Pressure in the receiver is indicated by a mercury manometer at C, and another manometer (not shown) gives the back pressure in the pipe J, which is always under a slight pressure due to friction. The latter manometer is used to assist in adjusting the difference of pressure on the orifice. No attempt is made to compute the evaporation in this manner. The moisture separated from the steam collects in the 3-in. trap pipe D, the amount being indicated in pounds and fractions on the gauge glass. Some moisture originally in the steam is of course evaporated in passing the reducing valve, consequently corrections of computed results for quality of steam and conversion to equivalent evaporation from and at 212° are made on the basis of the mean pressure maintained in the receiver—usually about 2 lbs.

Variation in load is obtained by the introduction of suitable orifices. The by-pass valve E allows changes to be made during operation, the orifices simply taking the place of a gasket in the union.

METHOD OF MAKING TESTS

The plan for the tests at Urbana was as follows: (1) The load to be maintained between 60 and 70% of the builders' rated capacity; (2) the load to be uniform throughout the test; (3) a steam pressure of 2 lbs. to be carried on the heating system; (4) a definite charge of coal to be sup-

plied of each firing; (5) each test to be of approximately 8 hrs. duration.

The standard method of starting and stopping the tests, as prescribed by the American Society of Mechanical Engineers, was used. A steam pressure of 5 lbs. was maintained in the boiler for 30 to 40 min. before starting. At the start the ashes and the partly burned coal were rapidly removed and a weighed amount of wood was ignited; this operation required about 3 min. during which the steam pressure dropped from 5 lbs. to 1 lb. As soon as the wood was lighted a signal was given to start the test and readings of time, water level, and pressures were taken. As soon as the wood was burning well a charge of about 25 lbs. of coal was fired. In 8 to 10 mins., when the coal was well ignited, the rest of a 75-lb. car of coal was fired. The fire was not touched during the remainder of the test, except at times when the steam pressure fell below 5 lbs. When this occurred the fire was poked and raked and another 75-lb. firing was made.

The following readings were taken every 15 min.: Height of water in boiler, height of water in tank, temperature of feed water, boiler-room temperature, steam pressure, pressure beyond reducing valve, back pressure in separator, height of water in separator, and drafts in ash pit, over the fire, and in the flue.

The air supply was taken entirely through the ash-pit door. On some of the tests it might have been advisable to admit a part of the air used for combustion continuously over the fire through the furnace door, or it might have been possible to increase the overall efficiency of the boiler by admitting air over the fire for a few minutes after each firing by cracking the furnace door; but as such a procedure is not commercially practicable, no attempt was made to prove its effect.

As the duration of the trials was to be approximately 8 hrs., just enough coal was put on at the last firing to keep up 5 lbs. steam pressure until time to close the test. When the steam pressure dropped below 5 lbs. on the last firing, the grate was dumped and the unburned coal was rapidly placed in a can having a tight-fitting cover which prevented further combustion. This unburned fuel removed at the close of the test, known as the residual fuel, was weighed separately from the ash that accumulated during the test. The ash and refuse from the coal were completely removed before the grates were dumped to clean them of the residual fuel. Just as the grates were dumped the final readings of time, water level, and pressures were taken.

The boiler flues were blown after the close of every test, so that the heat developed had an equal chance on the heating surface at the start

TABLE 6.—AVERAGE RESULTS OF TESTS ON STANDARD FUELS AT URBANA

BOILER A.

Kind of fuel.	Length of test (hours).	British thermal units per pound of dry fuel.	Efficiency (dry-fuel basis).	Cost of evaporating 1,000 pounds water from and at 212° F. (cents).[a]	CO_2 in dry flue gases (per cent).	CO in dry flue gases (per cent).	Flue temperature (° F.).	Per cent of volatile matter in the combustible.
Coke	16.83	12,805	55.39	7.31	4.9	0.38	651	3.30
Anthracite	16.46	13,208	52.20	7.52	4.1	1.70	589	8.35
Pocahontas	17.41	15,055	46.42	7.30	5.1	.47	607	20.99
BOILER B.								
Coke	15.24	12,916	62.14	6.38	13.2	0.25	480	2.65
Anthracite	17.17	13,205	61.34	6.50	9.2	.66	519	7.26
Pocahontas	18.21	15,055	58.52	5.92	8.2	.35	520	20.99

[a] Based on fuel at $1 per 2,000 pounds.

of each test; however, on some of the trials more soot was formed than on others, and the heating surface on these trials was therefore much less effective at the close than at the start.

Table 6 gives the average result of the tests at Urbana on three different kinds of standard fuel. Boiler B, a sectional boiler, shows a decided gain in efficiency when burning the same fuel over boiler A, a vertical boiler of the usual type. As the percentage of volatile matter in the combustion increases from 3.30 to

TABLE 7.—RELATION BETWEEN RATED CAPACITY DEVELOPED AND COEFFICIENCY WITH THREE DIFFERENT FUELS

Designation of fuel.	Test No.	Percentage of rated capacity developed.	Efficiency.
			Per cent.
Illinois No. 19 E (raw coal).	53	62.6	45.99
	55	58.7	49.25
	52	53.1	41.94
	54	47.3	38.67
Illinois No. 31 (briquets)...	34	62.9	51.88
	33	59.6	48.83
	44	58.2	41.94
Maryland No. 2 (briquets)..	21	69.1	56.13
	22	63.4	55.27
	23	60.6	49.29

20.99, the efficiency of boiler A drops about 9%., but that of boiler B drops only about 3.5%. This shows not only that boiler A is much less efficient than boiler B, but also that it is entirely unfitted for high volatile coals.

Table 7, compiled from four tests made on an Illinois coal raw, three tests on an Illinois coal briquetted, and three tests on a Maryland coal briquetted, shows the relation between capacity and efficiency of the three fuels.

Table 8 gives the relative values of several fuels for heating purposes. The results were obtained by evaporative tests on house heating boilers of the two different types—sectional and vertical. These tests were made at Urbana on boilers A and B. The chemical analysis, British thermal unit determination, the efficiency of the boiler are given for each fuel. The figures under relative fuel value show that with a sectional boiler the effectiveness of different fuels is in proportion to the number of ther-

TABLE 8.—RELATIVE VALUES OF FUELS FOR HOUSE HEATING BOILERS

Kind of fuel.	British thermal units.	Efficiency of boiler and grate.	Cost (in cents) of evaporating 1,000 pounds of water from and at 212° F. with price of fuel ranging from $1 to $10 per ton of 2,000 pounds.										Relative values of fuels (determined by evaporative tests) with price varying from $1 to $10 per ton of 2,000 pounds for Pocahontas coal.									
			$1.	$2.	$3.	$4.	$5.	$6.	$7.	$8.	$9.	$10.	$1.	$2.	$3.	$4.	$5.	$6.	$7.	$8.	$9.	$10.
SECTIONAL BOILER.		*Per ct.*																				
Pocahontas	14,753	55	5.92	11.84	17.76	23.68	29.60	35.52	41.44	47.36	53.28	59.20	$1.00	$2.00	$3.00	$4.00	$5.00	$6.00	$7.00	$8.00	$9.00	$10.00
	14,229	57	5.94	11.88	17.82	23.76	29.70	35.65	41.58	47.52	53.46	59.40	.997	1.994	2.991	3.988	4.985	5.982	6.979	7.976	8.973	9.970
Maryland	13,842	57	6.16	12.32	18.48	24.64	30.80	36.96	43.12	49.28	55.44	61.60	.961	1.922	2.883	3.844	4.805	5.766	6.727	7.688	8.649	9.610
Pennsylvania	13,640	57	6.25	12.50	18.75	25.00	31.25	37.50	43.75	50.00	56.25	62.50	.947	1.894	2.841	3.788	4.735	5.682	6.629	7.576	8.523	9.470
Do	12,376	61	6.38	12.76	19.14	25.52	31.90	38.28	44.66	51.04	57.42	63.80	.928	1.856	2.784	3.712	4.640	5.568	6.496	7.424	8.352	9.280
Coke	12,690	59	6.50	13.00	19.50	26.00	32.50	39.00	45.50	52.00	58.50	65.00	.911	1.822	2.733	3.644	4.555	5.466	6.377	7.288	8.199	9.110
Anthracite	12,690	42	10.00	20.00	30.00	40.00	50.00	60.00	70.00	80.00	90.00	100.00	.592	1.184	1.776	2.368	2.960	3.552	4.144	4.736	5.328	5.920
Illinois	11,501	42	10.93	21.86	32.79	43.72	54.65	65.58	76.51	87.44	98.37	109.30	.542	1.084	1.626	2.168	2.710	3.252	3.794	4.336	4.878	5.420
Do	10,535																					
VERTICAL BOILER.																						
Coke	12,055	55	7.19	14.38	21.57	28.76	35.95	43.14	50.33	57.52	64.71	71.90	.985	1.970	2.955	3.940	4.925	5.910	6.895	7.880	8.865	9.850
Pocahontas	14,753	45	7.30	14.60	21.90	28.20	36.50	43.80	51.10	58.40	65.70	73.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00
Anthracite	12,679	51	7.46	14.92	22.38	29.84	37.30	44.76	52.22	59.68	67.14	74.60	1.022	2.044	3.066	4.088	5.110	6.132	7.154	8.176	9.198	10.220

mal units they contain; thus, if all fuels were sold at the same price the cheapest would be the one containing the largest number of thermal units.

A part of this table is figured on the cost of evaporation 1,000 lbs. of water. Different fuels may be compared on this basis; for example, at a locality where anthracite costs $8 a ton and coke $6, the cost of evaporating 1,000 lbs. of water with anthracite in a sectional boiler would be $1.30 and $0.46, respectively, over the cost of Pocahontas at this particular locality. The relatively higher price of anthracite may be offset in public opinion by the fact that the coal is cleaner and burns without smoke.

At another place, should anthracite cost $8.20 a ton, Pocahontas coal would be worth $9 a ton, good Pennsylvania coal $8.65, coke $8.35, and good Illinois coal $5.33. If any of the four last-named fuels cost less

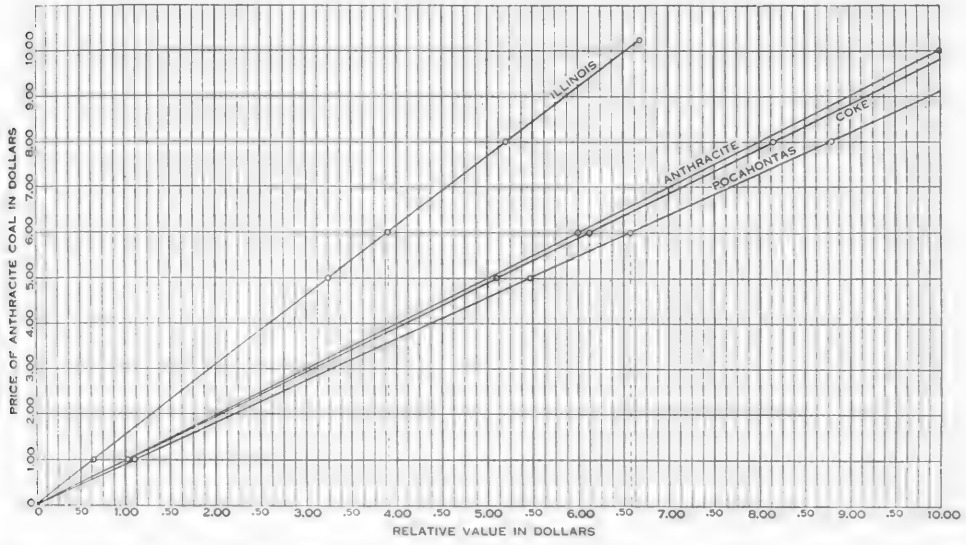


FIG. 5.—CHART SHOWING RELATIVE VALUE OF FUELS FOR HOUSE HEATING BOILERS

52 cents and with coke 38.28 cents. This means that at such a locality the same amount of work could be accomplished at a saving of about $2 a ton by using coke instead of anthracite. Similar comparisons of other fuels may be made from this table.

The last part of this table offers another method for determining the comparative values of fuel at any given locality. For instance, if at a certain place Pocahontas coal costs $5 a ton, anthracite is actually worth $4.56, coke $4.64, good Illinois coal $2.96, and Pennsylvania coal $4.80. If anthracite sells at this same place for $8, it is a luxury by $3.44 a ton over Pocahontas, which, on the other hand, may not be the cheapest available fuel in that market. If good Pennsylvania coal can be purchased for $3.50 a ton, or good Illinois coal for $2.50, they offer a saving of than the amounts stated, they would be just that much cheaper than anthracite coal.

A more convenient way of determining the comparative cheapness of different fuels is by means of a diagram like that forming Fig. 5, in which the cost of anthracite coal is taken as a standard, and the values of the other three fuels based on the results of tests are referred to it. Suppose a man wishes to know the relative values of Illinois and Pocahontas coals in a locality where anthracite can be had for $5 per ton. Placing a straight edge on the $5 horizontal line, he notes the point of intersection with the line showing the value of Illinois coal and the point at which a perpendicular from this point of intersection strikes the base and finds with anthracite at $5 a good Illinois coal is worth $3.25. In

the same way he finds that Pocahontas coal is worth $5.47. In another locality, where anthracite can be had for $6, Illinois coal, as shown by the diagram, is worth $3.90, Pocahontas coal $6.58, and coke $6.12.

In concluding, the author states that the results of these tests indicate that coals containing the higher percentages of fixed carbon give the least smoke and the best results for economy.

Design of Furnace Heating Systems, with an Application to a Ten-room Residence*

By Prof. J. D. Hoffman

The oxygen of the air fills about one-fifth of the volume in atmospheric air, and is the element that makes combustion possible. When rejected as a carbonic acid, CO_2, it carries away the impurities from the body and cleanses the system. The amount of CO_2 in the air is used as an index to the purity of the air, pure air having four parts CO_2 in 10,000 parts of air. The average person gives off probably .6 cubic foot of CO_2 in one hour, from which, if the impurity of the air in the room be maintained at six parts CO_2 in 10,000 parts of air, the person will need 3,000 cubic feet of atmospheric air per hour; if maintained at eight parts in 10,000, the person will need 1,500 cubic feet per hour. This may be put in a simple form for calculation:

Let V = cubic feet of atmospheric air needed per hour per person; A = cubic feet of CO_2 given off per hour per person; n = the standard of purity to be maintained (allowable parts of CO_2 in 10,000 parts of air); and p = the standard of purity in atmospheric air, say, 4; then

$$V = \frac{A}{n-p} \quad \ldots\ldots\ldots (1)$$

Eighteen hundred cubic feet per hour per person has become quite generally accepted, and will be assumed in this case. In addition to the vitiation of the air by the occupants, there are more or less impurities coming from gas burners, lamps, etc. This may be accounted for in terms of occupants of the room. It is claimed that one ordinary gas burner using, say, 4 cu. ft. of gas per hour will vitiate as much air as three or four people. Where many lamps or gas burners are used they should be accounted for by increasing the fresh-air supply accordingly.

AIR CIRCULATION IN FURNACE HEATING

The cubic feet of air per hour as a ventilator may be taken as 1,800 N where N is the number of persons to be provided for. For example, let t = temperature of the air entering the room from the register, say, 130° F.; t' = temperature of room, say, 70°, and t_0 = temperature of outside air, say, 0°. Let, also, H = heat loss from the room in B. T. U. per hour as calculated by formula. There is, of course, a certain loss of heat expected in any case by the air leakage. This is included in the value H, but by the B. T. U. carried away by the exhausted air is meant where special provision is made for venting to the outside air. The heat loss H will be the same for all methods of heating, and should be carefully worked out for the various rooms by the use of some well-tried formula, because all the other calculations in the system are based upon it.

It will be seen that the heat given off as the air drops in temperature from t to t', must equal the heat loss H. The safe temperature of the air as it leaves the register is 130° F. This may at times reach 140° F., but it is well to use the lower value for the calculations. If, as is usually the case, the room air is 70° F., the incoming air will drop in temperature through

*Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, in New York, January 19-21., 1909

60° F., and will give off 60 ÷ 55 = 1.09, say, 1.1 B. T. U. per cubic foot.

Since, in the design of any plant, H is the first thing to find, we will assume this value as having been obtained by some satisfactory formula, and we will base all the following results upon it. Having determined H, then the amount of air needed as a heat carrier measured at the register is,

$$Q = \frac{55\ H}{t - t'} = \frac{H}{1.1} \text{ approx.} \ldots\ldots (2)$$

where 55 cu. ft. of air may be heated from 59° to 60° F. by one B. T. U. This is proven as follows: The specific heat of air at 32° F. is .238, then one B. T. U. will raise 1 ÷ 0.238 = 4.2 pounds of air from 32° to 33°, and if one cubic foot of air at 32° weighs 0.807 pound, we have 4.2 ÷ 0.0807 = 52 cu. ft. of air at 32°, or 52 × 520 ÷ 492 = 55 cu. ft. of air from 59° to 60° F.

Formula 2 gives the required amount of air to carry the heat for ordinary building losses. This amount of air at the register will be practically constant for the chosen temperatures, no matter if the air be taken back to the furnace, *i. e.*, recirculated, or be vented to the air and fresh air taken in to fill its place. The only part of the plant that would be affected by this change from inside to outside air would be the size and capacity of the furnace.

IS THIS AMOUNT OF AIR SUFFICIENT FOR VENTILATION IF TAKEN FROM THE OUTSIDE?

Take the 13 × 15 × 10 ft. sitting room, Plate 2. Let the estimated heat loss be 14,000 B. T. U. per hour, then Q = 12,727 cu. ft. With a room volume of 1,950 cu. ft., the air will change 6.5 times per hour, and, allowing 1,800 cu. ft. of air per person, will supply seven people with good ventilation, if fresh air be used. Stated as a formula, this would be,

$$N = \frac{H}{1.1 \times 1800} = \frac{H}{2000} \text{ approx.} (3)$$

As a matter of fact, ventilation for half this number would be ample in an ordinary residence room excepting on extraordinary occasions. So it would seem that the subject of ventilating air will be more than taken care of if the ducts are planned to carry air for heating purpose only.

If for any reason Q is not sufficient for ventilation, then more air must be sent to the room and the temperature dropped correspondingly to avoid overheating the room. Let Q_v = total volume of air per hour, including extra air for ventilation, measured at the register, and t_v = temperature of the air at the register, then,

$$t' = 70 + (55\ H \div 2_v) \ldots (4)$$

HEAT LOST BY VENTILATION

Now, if this air which we call a heat carrier is not allowed to escape from the building through vent ducts, but is taken back to the furnace and recirculated, the only loss of heat will be that calculated by the formula. As a matter of fact, the air thus used would soon become contaminated and unfit for the occupants to breathe; hence, it is customary to exhaust through vent flues a part or all of the air sent from the furnace. All the air exhausted carries away an additional loss of heat from the building in proportion to the drop in degrees from 70 to that of the outside air. Take, for example, a room under three different conditions of service on a zero day: First, when all the air is recirculated; second, when only enough air is exhausted to give good ventilation; third, when all the air is exhausted.

Under the first case the loss will be the same as that calculated by the formula—*i. e.*, = H.

The second, assuming N people to need ventilation, will be,

$$H' = H = [(1800 \times N \times 70) + 55] \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

The third will be

$$H'' = [(t - t_0)\ H] \div (t - t') \ldots (6)$$

Formula 5 is a general formula, and will fit any condition and any degree of purity in ventilation. With N = 2 this would come near to average practice in the operation of furnace systems. Formula 6 is only true where the volume Q is circulated. If larger volumes are circulated, use Formula 5.

Little thought need be given, therefore, to the subject of ventilation in the average residence further than to plan for the entering of the fresh air to the furnace, and its withdrawal through vent ducts to the attic. Since the quantity of air is sufficient, the owner may regulate to suit his needs.

Both lines of air entering the furnace should be dampered. Concerning the best point from which to return the room air, nothing very definite can be stated. It is best taken from the coldest part of the house, generally near a door.

MOISTURE IN AIR

Moisture in the air is a benefit both to the heating and the ventilating systems in any room. With moist air in the room, a person may feel comfortable when the temperature is several degrees lower than the comfortable temperature of dry air. Dry air absorbs the moisture from the skin. The vaporization of this moisture causes a loss of heat from the body, and gives to the person a sense of cold, which is only relieved when the temperature of the room is increased. Air that is fairly saturated with moisture will not permit of much evaporation from the skin, because there is not much demand for this moisture in the air; consequently, the body retains that heat and the person has a sensation of warmth which is only relieved by lowering the temperature of the air in the room. On the other hand, at low temperature the moisture in the air chills the surface of the skin by convection, a condition that is not so noticeable when the air is dry. It follows from the above statement that the range of comfortable temperatures is less for moist air than for dry air.

Concerning the effect of moisture in the air in its relation to the heating and ventilating of the room; dry air does not absorb radiant heat; moisture in the air has this quality. It has also somewhat less weight than the dry air and is more buoyant. Because of the possibility of storing up the radiant heat within the particles of moisture, and because of its convection qualities, it serves as a good heat carrier for the heating system.

HUMIDITY OF THE AIR

The *actual humidity* of the air is the amount of moisture in the air at any temperature. The *relative humidity* is the amount of moisture actually in the air divided by the amount of moisture which the same volume of air could hold at the same temperature when saturated. It is very important that the heating engineer be able to add to or to take away from the amount of moisture in the air supply of any building. To find the amount of moisture that should be added or subtracted in any case, it is first necessary to determine the humidity of the air at various points along its course. This may be obtained by the aid of the wet-and-dry-bulb thermometer, or by any one of a number of hygrometers supplied by the trade.

THE THEORETICAL AMOUNT OF MOISTURE TO BE ADDED TO AIR SO AS TO MAINTAIN A CERTAIN HUMIDITY

Warm air has a much greater capacity for holding moisture than cold air. When air is taken at a given outside temperature and heated for interior service, the volume increases with the absolute temperature and the humidity decreases rapidly. Air thus treated becomes dry and unpleasant to the occupant, as well as being detrimental to the furnishings of the room. Some means should, therefore, be provided to supply this moisture to the air current.

In calculating the amount to be added, let Q = volume of air in cubic feet per hour entering the room at the register; t = temperature in degrees F., and T = its absolute temperature; let Q' and Q_0 = the corresponding volumes of air in the room and on the outside, with t' and t_0 their temperatures in degrees F., and T' and T_0 = their absolute temperature; also, let h' and h_0 be the humidities, respectively, of the room air and the outside air. Then, from the formulas, $TQ' = T'Q$ and $TQ_0 = T_0Q$ . . . . (7) find Q' and Q_0.

From prepared tables find the amounts of moisture M' and M_0 in 1 cu. ft. of saturated air at the tem-

peratures t' and t_0; multiply these by the respective humidities and volumes, and the difference between the two final quantities will be the amount of moisture required per hour as expressed by the formula, .

$$W = Q' M' h' - Q_0 M_0 h_0 \ldots\ldots (8)$$

Let $Q = 5000$, $t = 130$, $t' = 70$, $t_0 = 30$, $h' = .50$, $h_0 = .50$, $M' = 7.98$, and $M_0 = 1.958$, then

$Q' = (5000 \times 530) \div 590 = 4500$.

$Q_0 = (5000 \times 490) \div 590 = 4150$.

$W = 13,890$ grains, or 1.984 lbs. per hour.

This means that approximately 2 lbs. of water would be evaporated for every 5,000 cu. ft. of fresh air entering the register.

The practical difficulties experienced in attempting to moisten air that is being raised to a higher temperature are great. When a current of air passes over the surface of still water, as is the case with moistening pans, it takes up very little of the water vapor unless both the air and the water are at a fairly high temperature. If the water is in motion, as produced by a spray or a drip, then the air will take up large quantities of the moisture, but much of it will be merely suspended in the air in a very finely divided state and will be deposited on the first cool object it comes in contact with. Such moisture cannot properly be said to be *in* the air.

HEAT REGISTERS

The velocity of the air as it leaves the register is best taken between 3 and 4 ft. per second to insure against unpleasant drafts in the room. This is especially true for registers located near the floor line. Assuming the mean figure 3.5 ft. per second, we have.

Area of net heat register $= 55\ H \times 144 \div (60 \times 3.5 \times 3600)$
$= 0.01\ H$ square inch, approx. .....(9)

The *nominal size* or the *catalogue size* of the rigster is about,

Gross heat register $= 1.5 \times$ net heat register .....(10)

Floor registers are best adapted to halls and rooms on the first floor. There is a serious objection to their use, however, when they require the cutting of carpets or hardwood floors. This restricts their use very much. For rooms on the second floor the wall register is preferred. Floor registers on the first floor may safely be placed on the more exposed sides of a room when this can be done without making the leaders too long. Wall registers should always be in the inner walls.

HEAT STACK

To get the proper size of the stacks in any heating system is a very important part of the design of that system. By some designers the cross-sectional area is taken roughly as a certain ratio to that of the net register. This has been quoted anywhere from 50% to 90%. Such wide variations between extremes of air velocity should certainly receive careful application. Professor Carpenter suggests 4, 5 and 6 ft. per second, respectively, as the air velocity for first, second and third floors. From the basis of the net register (figured at 3.5 ft. per second) two quotations give stack values approximately as follows: First floor, 80% to 88% of the net register; second floor, 55% to 70%, and the third floor, 44% to 60%. In any case, the formula for the area of the stack is developed in the same way as the net register by substituting the approved air velocity in the place of 3.5.

The air velocity in the stack is based upon the formula $V = \sqrt{2gh}$, where $h = (\text{effective height of stack}) \times (t - t') \div (460 + t')$. V is in feet per second; t is the temperature of the stack air, and t' is the temperature of the room air. The calculated results from this formula are much higher than obtain in practice because of the shape of cross sections of the stack, the friction of its sides and the abrupt turns in it. Stack values may be safely taken as follows:

First floor, S. A. (sq. in.) .007 H
Second floor, S. A. (sq. in.) .006 H .....(11)
Third floor, S. A. (sq. in.) .005 H

Good-sized stacks are always advisable, but because of the limited space between the studding it becomes necessary at times to put in a stack

that is too small for the room or to increase the thickness of the wall. *Most second-floor stacks are too small* to heat the rooms properly. The shape is also against maximum efficiency. Round sections are best, but these are difficult to place. Deep rectangular sections, such as 5½ in. by 13½ in., work well; but the shallow sections that fit in a 4-in. studded wall should be applied to only the smallest rooms.

Results from different installations show that there is a variation in the practice of proportioning stacks, as commonly installed, of 50% or more. This is not so noticeable in the first-floor rooms when stacks are used as it is in those of the second floor. Some of this difference is no doubt due to the fact that the thickness of the walls has been previously determined by the architect, and since the heating system was probably a secondary consideration, no party would take the initiative in changing the sizes. In many cases, too, the owner is indifferent or possibly advocates small stacks because of the use to which the room is to be put, *i. e.,* less heat for sleeping rooms, etc., etc.

A good rule, and an old one, is to heat every room as though it were to be a living room in the coldest weather, and then let the owner regulate to suit his needs, even to a complete closure of the duct. Where thin partition walls are put in, say, 4-in. studding set 16-in. centers, and not enough space is allowed for a single stack, two stacks may be put in. This is somewhat more expensive, but it is very satisfactory, since it gives a better distribution of heat in the room, and when all the heat is not needed, as in mild weather, one line may be shut off entirely.

A better way is for the architect to anticipate this state of affairs and specify walls of sufficient size. The ideal conditions will be reached when the architect actually provides air shafts of sufficient size to accommodate either a round or a nearly square stack. When this time comes, a great many of the furnace-heating difficulties will have been solved. This point has many times been well shown where furnace systems have been installed in old houses. Large rectangular shafts or stacks were run up the corners of first-floor rooms to reach second-floor rooms. These second-floor rooms always had heat to spare. Stacks placed in this way and finished over to suit the finish of the rooms are not very unsightly; and how much better, still, were they planned by the architect and properly enclosed so as to harmonize in the general layout of the room!

All stacks should be located in inside or partition walls and, if possible, should look toward an outside or cold side of the rooms. This protects the air current from excessive loss of heat, as would be the case in an outside wall. It also provides more uniform distribution of air. Two rooms should not be taken from the same stack, but two stacks may run to the same room and may be taken from the same leader.

VENT REGISTERS AND STACKS

It is well to install vent registers and stacks in the inner walls leading to a general vent in the attic. If all the air were to pass out of the vent register at the same velocity as it entered through the heat register, the area of the vent register would be to the area of the heat register as the ratio of the absolute temperatures of the leaving and entering air. With $t' = 70$ and $t = 130$, we have theoretically

Net vent register = (530 ÷ 590 = 0.9) net heat register.

As a matter of fact, however, some of the air leaves through other openings, and the vent registers need not be so large. The following will be found satisfactory:

Area vent register = .8 area heat register ..... (12)

also,

Area vent stack = .8 area heat stack ..... (13)

LEADER PIPES

Since all the air that passes through the stacks must pass through the leader pipes, it would at first seem reasonable to assume that the areas of the two would be equal. It must be remembered, however, that the stacks, because of their vertical position, have

less friction than the nearly horizontal leader pipes, and will consequently retard the air less. There will also be some loss of heat in the air as it passes through the leader pipes; consequently, the volume of air entering the leader from the furnace will be greater than that going up the stack. It would be well from the above reasons to make,

Area of leader pipe = (1.1 to 1.2) stack area .....(14)

the exact figures to depend upon the length and inclination of the leader and the selection of the diameter of the pipe. All leaders should be round and should be free from unnecessary turns. They should be given a good upward pitch toward the stack, and should be covered to avoid heat loss and to provide safety to the plant. The connection between the leader and the stack should be by a long radius bend, and should not head into the stack at right angles.

COLD AIR DUCT

The area of the cold air duct is determined largelv by experience, as in the case of the vent register. It may be,

Area cold air duct = 0.8 total area of leaders .....(15)

Assume the average velocity of the air in the leaders to be 6 ft. per second, and the area of the cold air duct to be as shown above; then, if the air in each were of the same temperature, the velocity in the cold air duct would be $6 \div 0.8 = 7.5$ ft. per second, but since the temperatures are different, the velocities will be in proportion to the absolute temperatures. Hence, it is at 0° F., $0.78 \times 7.5 = 5.8$; at 25° F., $0.82 \times 7.5 = 6.2$, and at 50° F., $0.88 \times 7.5 = 6.6$ ft. per second. It is seen by this that although the area of the cold air duct is contracted to 80% of that of the leaders, yet the velocity in all cases is below that of the leaders.

It is always well to have a cold air duct that is large in cross-sectional area and free from obstructions and sharp turns. The air should be taken from a point on the outside as high above ground as convenient, and this opening should be vertical and well screened. The inside ducts should be tight, and should have no drainage water or refuse collecting in it. If an inside return circulation be added, both lines may unite near the furnace, and each line should be valved.

If it is decided that there never will be a time when all the air Q need be taken from the outside, then the area of the cold air duct may be reduced a certain amount. This is not a very safe policy, and should be done only when conditions seem to demand it.

The recirculating duct and register will be proportionted to the cold air line to suit each individual case. The best size, of course, is that of the cold air line.

THE FURNACE

A good way to select the furnace size is by grate area, and this is estimated from the total heat loss from the building, figured on the basis of a certain degree of ventilation. In obtaining the grate area, it is necessary to assume the quality of the coal, the efficiency of the furnace and the pounds of coal burned per hour per square foot of grate.

The quality of the coal selected would be between 13,000 and 14,000 B. T. U. per pound, the efficiency of the average furnace is about 60%, and the coal burned per square foot of grate per hour ranges from 3 to 7 lbs. Concerning the last point, there may be a wide difference of opinion. Higher temperatures in the combustion chamber are conducive to economy, hence to reduce the size of the firepot and the combustion chamber and firelight often would seem to be the ideal way.

On the other hand, with high temperatures in the combustion chamber, the loss up the chimney is increased. Probably the one factor which is most effective in settling this point is the inconvenience of frequent firing. Furnaces are charged from two to four times every twenty-four hours. This requires a good-sized firepot and a possibility of banking fires. To allow 5 lbs. per hour is probably as good an average as can be made for most coals in furnace work.

(To Be Concluded in the April Issue.)

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IN ORDER that there may be no doubt in the minds of any of our readers as to the course adopted by the American Society of Heating and Ventilating Engineers regarding the rating of house heating boilers, we would call renewed attention to the formal actions taken at the meeting. The first action was the passage of a resolution introduced by Mr. William C. Vrooman, as follows:

"Resolved, that it is the sense of this society that, in the absence of any standard for rating steam and hot water heating boilers, one should be established, based on coal consumed per square foot of grate surface and efficiency."

This was the only action taken that placed the society definitely on record in favor of making coal consumption and efficiency a basis of boiler rating.

Another resolution, amplifying the one given above, was introduced by Mr. Percival H. Seward and supported by Prof. R. C. Carpenter, which, after much favorable discussion, was referred to the Committee on Standards. The text of this resolution was published in the February issue.

Thus, while the society's action commits it to a definite basis of rating, the details of such a rating will depend on the further action of the Committee of Standards.

AN AUTHORITATIVE statement on the economy of the use of briquets in house heating boilers is made in the government report of the recent tests of fuels at St. Louis and Urbana, Ill. A careful comparison of the results of the tests on the coal and the briquets seems to show no advantage in the briquets over coal of a suitable size for house heating boilers. Briquetting a good bituminous coal, according to the experts, would be justified only when slack is used for materials and the gain from briquetting would lie almost entirely in the more favorable size of the fuel. This gain would be less for coals that coke readily than for noncoking coals or for coking coals that are not suitable for domestic purposes in the form of slack. Briquets from such coal burn fairly well, as they allow the air to pass up through the fuel bed.

The experiments showed that the pitch binders used are not suitable for a furnace working at the low temperatures common in a house-heating boiler, as they volatilized and in most cases escaped unburned or were deposited on the surface of the boiler. In the St. Louis tests this coating burned off once or twice a day, causing a high temperature in the flue and, as a consequence, danger from fire.

Cost of Heat from Central Plants

In connection with the recent Government tests regarding the value of various fuels for house heating boilers, a general inquiry was made in the matter of central heating plants. The results obtained in this inquiry, when compared with the results of the tests on house-heating boilers, show that such plants are economical for the house-holder, and that they solve a part of the smoke problem in the cities.

Inquiries were sent to all the known commercial steam or hot-water heating plants of the United States. This was done because the substitution of large units equipped for burning coal in the most economical manner in the place of small units of less efficient design offers one way of reducing the total quantity of smoke given off in burning a certain amount of coal, and in addition is a possible factor in conserving the nation's fuel supplies.

Inquiries were sent to 150 plants and replies were received from 57. The information gathered from these replies is to be published in detail in a bulletin now in preparation. Of the plants that answered 41 gave figures showing the price of heat to the consumer either per square foot of radiating surface or per 1,000 lbs. of condensed steam. The figures are presented in the accompanying table.

On a comparison of the figures showing the rate per 1,000 lbs. of condensed steam with those of cost for evaporating 1,000 lbs. of water given in Table 8 (Page 22 of this issue), it becomes evident that considering all the factors concerned in the production of heat on the premises, such as cost of plant, depreciation, repairs, cost of fuel and labor, and the elimination of all discomfort from dust, ashes, or smoke, the house-holder is fortunate who can purchase heat from a central station at a reasonable price.

TABLE SHOWING PRICE OF HEAT FROM CENTRAL HEATING PLANTS

State.	Steam.		Hot water.
	Per square foot.	Per 1,000 pounds.	Per square foot.
	Cents.	*Cents.*	*Cents.*
Colorado	65		
Illinois	24		
Do		45	
Do	25		
Do	28		
Do	25		15
Do	22.5		17.5
Do			15
Do			20
Indiana			20
Do			18
Do			15.5
Do			12.5
Do			15
Do			17
Iowa			18
Do			20
Do			15
Do		50	
Minnesota		60	
Missouri		56	
Missouri			25
Montana	60		
New York		50	
Do		50	
Do		42.5	
Do		48	
North Dakota		60	
Do		40	
Ohio			15
Do			15
Do			20
Do			17.5
Do		50	
Pennsylvania	25		
Do		40	
Do	34		
Do	33.3		
Rhode Island		66	
Wisconsin			25
Do			25

Economy of Vacuum Exhaust Steam Heating as Shown by Indicator Cards

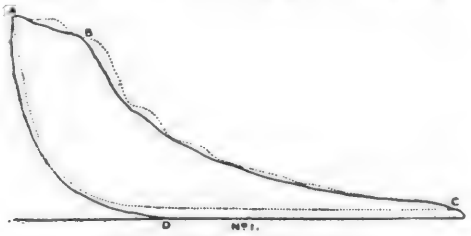
A good instance of the way in which the economical operation of the power plant as a whole can be sacrificed to increase the efficiency of one of its component parts is found in the common practice of using live steam for heating purposes while operating the engine condensing.

When water for condensing purposes can be obtained cheaply a saving of about 20% in the steam consumption of the engines can be made by the use of a condenser, but when a heating apparatus is to be supplied at the same time, it is more economical to shut down the condenser, and utilize the exhause steam for heating purposes even though the engine uses a little more steam per horse power.

Assuming the engine indicates 500 H. P. and uses 20 lbs. of steam per horse power non-condensing or 16 lbs. of steam per horse power condensing, and that there are 30,000 sq. ft. of radiation requiring about 9,000 lbs. of steam per hour, when operating condensing the engine uses 8,000 lbs. of steam per hour which, added to the 9,000 lbs. of steam needed for heating apparatus, makes 17,000 lbs.

of steam, equivalent to 567 boiler horse power.

If the condenser is shut down and the exhaust utilized for heating purposes, the engine will use 10,000 lbs. of steam, of which 9,000 lbs. as exhaust is used for heating purposes and 1,000 lbs. available for heating feed water and other purposes



INDICATOR CARD NO. 1

—the boiler horse power needed in this case being 333—a reduction of 234 H.P.

Assuming coal at $2.00 per ton and an evaporation of 7 lbs. of water per pound of coal, both of which are low, the increased cost of using high pressure steam for heating while operating a condenser amounts to $2,000 per heating season. The actual saving is greater than this, however, because when the condenser is operating, the feed water is at a much lower temperature than the hot water of condensation returned from the heating apparatus. The saving from this source is usually about 10%.

From what has been stated above, it is evident that the use of exhaust steam for heating purposes is essential to any real economy of operation.

Exhaust steam cannot however be successful circulated by pressure. The only practical method of insuring a positive circulation regardless of the distance which the steam must travel, is by installing a system of vacuum heating; an inspection of the indicator cards reproduced herewith shows why this is so.

The cards are so drawn that the motion of the engine piston is reproduced by the horizontal motion of the indicator pencil, while the pressure in the cylinder, corresponding to each position of the piston is represented to scale by the vertical motion of the pencil. The area of the card is proportional to the work done by the engine.

Steam is admitted to the cylinder while the piston moves from A to B, is cut off at that point, and expands while the piston passes from B to C. At C the exhaust valve opens and remains open while the piston returns to D, where it closes; and the steam is compressed as the piston returns to the starting point.

The two cards represent practically the same power, but the dotted card was taken when the engine was exhausting against a back pressure of 4 lbs. to force steam through a heating system.

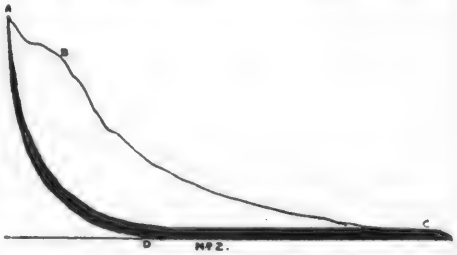
These cards were taken under ordinary working conditions and the one without back pressure shows about 5% more power than the dotted card. Comparison of the steam lines A-B shows that the cut off was 25% longer when working at a back pressure. That is, the engine used 25% more steam to produce 5% less power. If the power delivered in each case had been the same, the difference in cut off would have been more than 25%.

The black area on card No. 2 shows the power which is lost on account of back pressure of 4½ lbs. In this case it is 26.9% of the available power of the engine at this cut off. The application of a vacuum system would add this 26.9½ to the power of the engine without increasing the fuel bill.

This being true, it follows that but 500 boiler horse power would be required for the above plant, and the saving in the first cost of boilers alone would be enough to pay for a vacuum system twice over.

It is possible for a plant composed of engine, boilers, etc., of the highest grade to show a low commercial efficiency where exhaust steam is used for heating, and a varying back pressure is maintained on the engine.

The exhaust valve on a high grade engine designed for use without back pressure do not operate as well as when running at or near atmospheric pressure, and the steam valves are often forced to



INDICATOR CARD NO. 2

operate under unfavorable conditions, owing to the abnormally long cut-off required to keep up the speed.

When a system of vacuum heating is used, however, the utilization of the exhaust steam does not place any back pressure on the engine, and therefore overcomes this objection.—*From the new catalogue of the Consolidated Engineering Co., Chicago.*

Sirocco Engineering Works, Belfast, Ireland, represented by S. C. Davidson, has won its suit against the Serlano fan and an injunction has been granted against the defendants. The suit was practically undefended. The result of this suit, following the same outcome of similar suits in the German courts, is especially interesting in view of the recent consolidation of the Sirocco Engineering Co. with the American Blower Co.

Current Heating and Ventilating Literature

Under this heading is published each month an index of the important articles on the subject of heating and ventilation that have appeared in the columns of our contemporaries. Copies of any of the journals containing the articles mentioned may be obtained from THE HEATING AND VENTILATING MAGAZINE *on receipt of the stated price.*

CAR HEATING

Vapor Car Heating System. Illustrated description of the system invented by Edward E. Gold. 2,200 w. Ry & Loc Engng—Jan., 1909. 40c.

FAN HEATING

Heating and Ventilation (Fan and Steam-Coil) of the Union Terminal Station at Washington, D. C. Eng Rec.—Jan. 2, 1909. 2 figs. 3,500 w. 20c.

FEED-WATER HEATING

Means and Methods for Heating the Feed-Water of Steam Boilers. Reginald Pelham Bolton. This third and last article of a series examines the economy of heating feed-water by exhaust steam. Ills. 1,600 w. Engineering Magazine—Feb., 1909. 40c.

Relative Rate of Heat Transfer to Water At and Below the Boiling Point. W. M. Sawdon. Reports tests made at Sibley College laboratory. 1,500 w. Power—Jan. 12, 1909. 20c.

HEAT INSULATION

The Thermal Conductivity of Heat-Insulators. A review of the research work of Dr. Wilhelm Nusselt. 3,300 w. Engng—Jan. 1, 1909. 40c.

HOT WATER HEATING

The design of Low-Pressure Hot Water Heating Systems (Calcul des Installations de Chauffage par l'eau Chaude à Basse Pression). E. Mathieu. A mathematical discussion. Ills. Serial.

Fourth part, 1,000 words. September, 1908. 40c.

Fifth part, 1,500 words. November, 1908. 40c.

Sixth part, 2,800 words. December, 1908. 40c.

Seventh part, 3,200 words. January, 1909. 40c.

Eighth part, 3,600 words. January, 25, 1909. 40c.

Ninth part, 3,000 words. February, 10, 1909. 40c.

PIPE STRENGTH

The Collapse of Tubes Under External Pressure. S. E. Slocum. A study of recent experiments. 3,500 w. Engng—Jan. 8, 1909. 40c.

VENTILATION

Ventilation of Three Basement Floors of the Marshall Field Retail Store, Chicago. Illustrated description of method adopted for ventilating three stories below the street level. 2,000 w. Eng Rec—Jan. 23, 1909. 20c.

The Weather for February

February, 1909, will go down in the weather records as one of the mildest Februarys this country has ever experienced. In all cities from which reports were received the mean temperature ran from 1° to 5° above the mean of this month for 39 years. These figures, taken with those for the previous months of the winter of 1908-1909 are so exceptional as to have an important bearing on the winter temperature averages in the cities in question.

NEW YORK

Highest temperature, 58° on the 10th; lowest temperature, 5° on the first; greatest daily range, 25° on the 10th; least daily range, 6° on the 5th. Mean temperature for the month 37°, 6° above the mean of this month for 39 years. Total precipitation, 4.31 in.; snowfall, 1.4 in.; no snow on ground at end of month. The precipitation for the month was a little over ½ in. more than usual. Prevailing direction of wind, west; total movement, 11,012 miles; average hourly velocity, 16.4 miles. Three days were clear, 11 partly cloudy and 14 cloudy. Rain fell on 12 days.

BOSTON

Highest temperature, 59° on the 5th; lowest temperature, 3° on the 1st; greatest daily range, 32° on the 10th; least daily range, 5° on the 7th. Mean temperature for the month, 33°, 5° above the mean of this month for 39 years. Total precipitation for month, 4.71 in., nearly ¾ in. more than usual; snowfall, 2,3 in.; trace of snow on ground at end of month. Prevailing direction of wind, west; total movement, 8,575 miles; average hourly velocity, 12.8 miles. Eight days were clear, 9 partly cloudy and 11 cloudy. Rain fell on 10 days.

CHICAGO

Highest temperature, 57° on the 4th; lowest temperature, 12° on the 25th; greatest daily range, 32° on the 9th; least daily range, 4° on the 15th. Mean temperature for the month, 32°, 5° above the mean of this month for 39 years. Total precipitation for month, 3.84 in., 1½ in. more than usual. Snowfall, 10.1 in.; trace of snow on ground at end of month. Prevailing direction of wind, southwest; total movement, 11,644 miles; average hourly velocity, 17 miles. Four days were clear, 3 partly cloudy and 21 cloudy. Rain fell on 11 days.

PITTSBURG

Highest temperature, 64° on the 24th; lowest temperature, 9° on the 1st; greatest daily range, 36° on the 24th; least daily range, 4° on the 28th. Mean temperature for the month, 37°, 5° above the

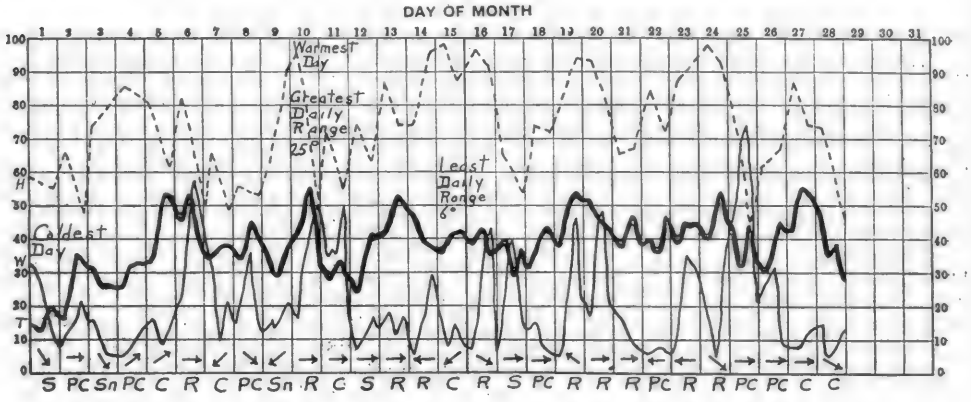
mean of this month for 39 years. Total precipitation for month, 4.47 in., over 1½ in. more than usual. Snowfall, 0.5 in.; trace of snow on ground at end of month. Prevailing direction of wind, west; total movement, 8,789 miles; average hourly velocity, 13.1 miles. Four days were clear, 7 partly cloudy and 17 cloudy. Rain fell on 16 days.

ST. LOUIS

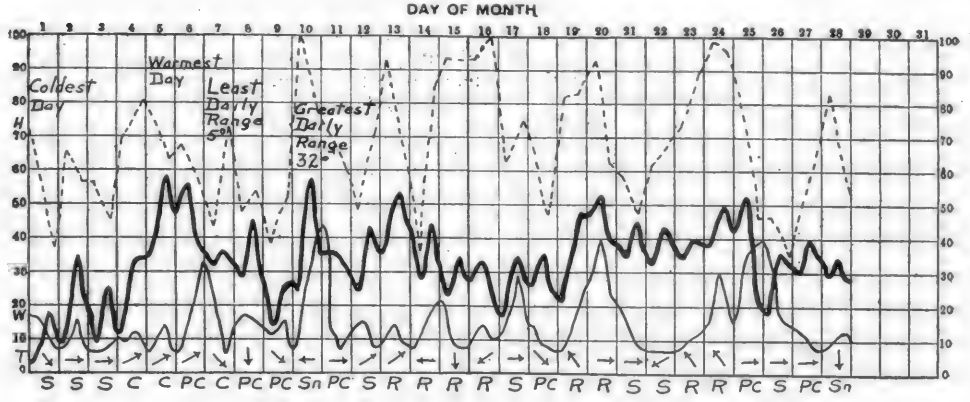
Highest temperature, 65° on the 4th; lowest temperature, 13° on the 1st; greatest daily range, 37° on the 9th; least daily range, 5° on the 12th. Mean temperature for the month, 39°, 5° above the mean of this month for 36 years. Total precipitation, 3.94 in., over 1 in. more than usual. Snowfall, 4.7 in.; no snow on ground at end of month. Prevailing direction of wind, south; total movement, 8,955 miles; average hourly velocity, 13.3 miles. Nine days were clear, 7 partly cloudy and 12 cloudy. Rain fell on 13 days.

To Eliminate Fog in a 60-Foot Radius

All navigators will be interested in the news which comes from Liverpool that Sir Oliver Lodge recently succeeded in completely clearing away for a radius of 60 ft. a thick fog by means of electrical discharges. The Marine Journal states that the Lodge system consists of driving electricity into a fog-laden atmosphere at a very high voltage from a series of disks at the tops of poles. The current acts directly upon the constituents of the fog, the electric influence whirling these hither and thither, causing them gradually to fall to the ground. A test is soon to be made of the apparatus in London.


DAY OF MONTH
Warmest Day
Greatest Daily Range 25°
Least Daily Range 6°
Coldest Day
H
W
T
S PC Sn PC C R C PC Sn R C S R R C R S PC R R R PC R R PC PC C C

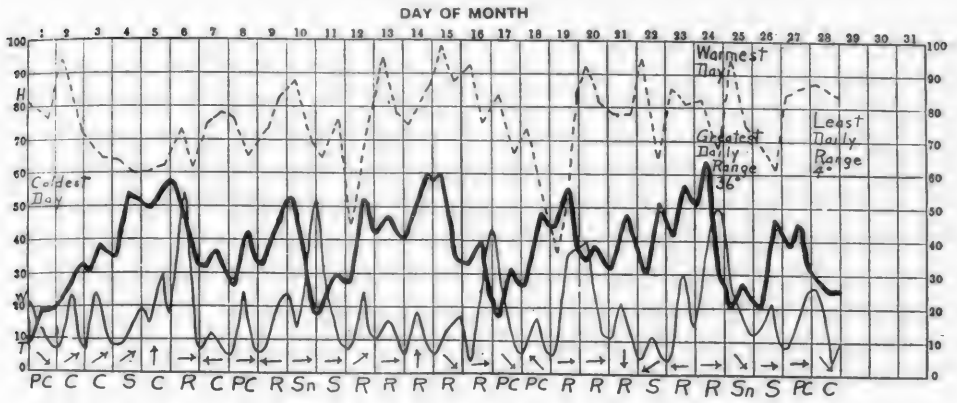
RECORD OF THE WEATHER IN NEW YORK FOR FEBRUARY, 1909


DAY OF MONTH
Coldest Day
Warmest Day
Least Daily Range 6°
Greatest Daily Range 32°
H
W
T
S S S C C PC C PC PC Sn PC S R R R R S PC R R S S R R Pc S PC Sn

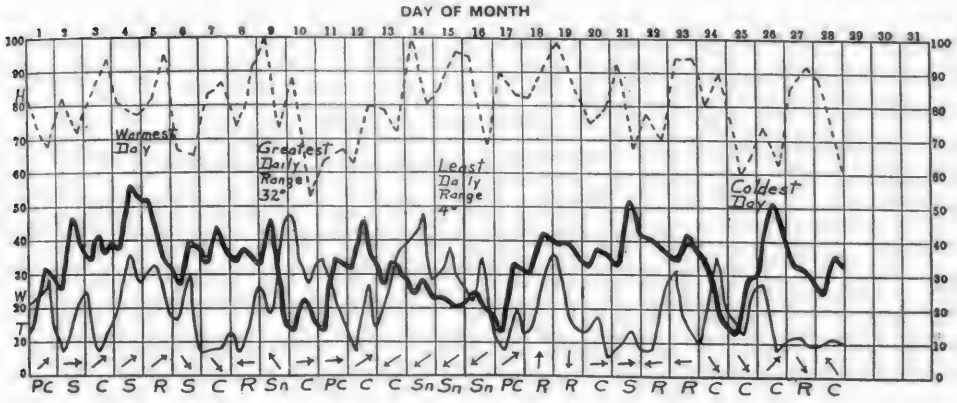
RECORD OF THE WEATHER IN BOSTON FOR FEBRUARY, 1909

Plotted from records especially compiled for THE HEATING AND VENTILATING MAGAZINE. by the United States Weather Bureau

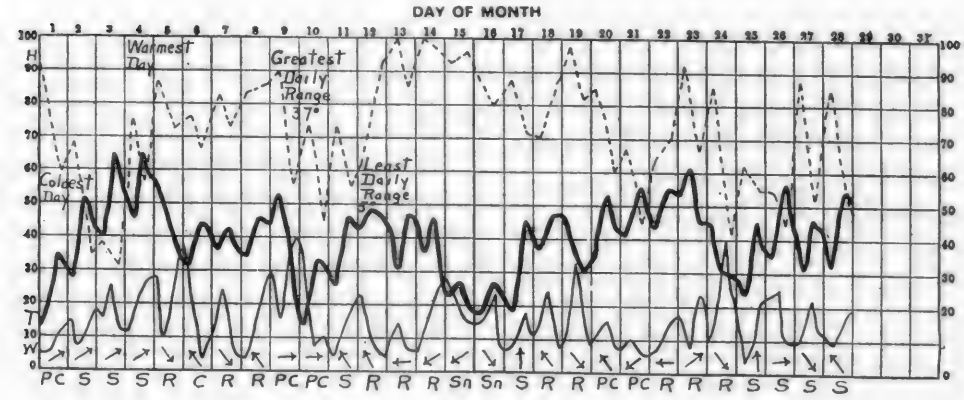
Upper line indicates temperature in degrees Fahr.
Broken lines indicates relative humidity in percentage from readings taken at 8 a. m. and 8 p. m
Lower line indicates velocity of wind in miles per hour.
Arrows fly with prevailing direction of wind.
S—Clear. PC—Partly Cloudy. C—Cloudy. R—Rain. Sn—Snow.


DAY OF MONTH
Warmest Day
Greatest Daily Range 36°
Least Daily Range 4°
Coldest Day
PC C C S C R C PC R Sn S R R R R R PC PC R R R S R R Sn S PC C

RECORD OF THE WEATHER IN PITTSBURG FOR FEBRUARY, 1909


DAY OF MONTH
Warmest Day
Greatest Daily Range 32°
Least Daily Range 4°
Coldest Day
PC S C S R S C R Sn C PC C C Sn Sn Sn PC R R C S R R C C C R C

RECORD OF THE WEATHER IN CHICAGO FOR FEBRUARY, 1909


DAY OF MONTH
Warmest Day
Greatest Daily Range 37°
Least Daily Range 3°
Coldest Day
PC S S S R C R R PC PC S R R R Sn Sn S R R PC PC R R R S S S S

RECORD OF THE WEATHER IN ST. LOUIS FOR FEBRUARY, 1909

Plotted from records especially compiled for THE HEATING AND VENTILATING MAGAZINE, by the United States Weather Bureau

Upper line indicates temperature in degrees Fahr.
Broken lines indicates relative humidity in percentage from readings taken at 8 a. m. and 8 p. m.
Lower line indicates velocity of wind in miles per hour.
Arrows fly with prevailing direction of wind.
S—Clear. PC—Partly Cloudy. C—Cloudy. R—Rain. Sn—Snow.

CORRESPONDENCE

Firing of Boilers During Test

EDITOR HEATING AND VENTILATING MAGAZINE:

In an article in your February issue (page 24, second col.) entitled, "A Standard Rating of Steam and Hot Water Heating Boilers," there is a statement which is not generally correct, and, I believe, may lead to confusion, if not corrected.

Mr. Seward says: "We arrive at the total consumption of about 40 lbs. for each square foot of grate for one firing. This is the amount of coal to be actually consumed; therefore, we must add to it sufficient for a bed on which to ignite the new charge, such surplus usually being estimated as one-fourth of the amount to be consumed or, in this case, 10 lbs., making the total charge of 50 lbs. for each square foot of grate. This charge of 50 lbs. to be handled each 8, 7 or 6 hours . . . figures up to 250 lbs. or one-eighth of a ton as required for one charge on a boiler of 5 sq. ft. grate area," etc.

The above statement is true only for the first firing when starting with a clean grate since, in subsequent firings, the grate would already have the bedding charge and would require only 40 lbs. per square foot or 200 lbs. for one firing, instead of 250 lbs.

M. F. THOMAS.

Ithaca, N. Y., March 4, 1909.

On the Prospective Rating of Boilers

EDITOR HEATING AND VENTILATING MAGAZINE:

The writer has noted with interest the further discussion of this subject, especially as carried on at the late convention of the heating and ventilating engineers. It is at least encouraging to know that disinterested authority in this relation is actively awake to the unsatisfactory condition of affairs in connection with manufacturers' claims as to the capacity of their boilers.

The proposed methods for a disinterested rating of boilers are all right so far as they go; but unless the matter is carried very much further by careful safeguarding of the testing, no satisfactory settlement of the question will have been attained. No system of tests will be satisfactory that is not entirely devoid of such influence as may be brought into action by the expert manipulation of the maker and seller of the appliance. Unless one has personally undertaken this matter of testing boilers, no one would believe how even the most honest and capable individuals making the test may be self-deceived as to the results, or how the tests can be influenced by expert stoking, etc. No two tests of the same boiler, even if conducted by the same individual, ever give the same results, owing to the various atmospheric and other conditions continually changing. Even with fuel of as like a quality as it is possible to secure, no two charges of the fuel ever produce exactly the same results. Nor does the same boiler ever produce exactly the same results under different conditions of installation.

The only approximation to a satisfactory outcome in testing would be in having boilers tested comparatively, at the same place, by the same manipulator, and under practically equal working conditions, a number of times repeated. It would be totally unfair as to the result of the test to allow the possibility of expert manipulation of the test by an interested person to be assumed to be scientifically correct. A very good boiler would make a very poor showing if poorly handled; and a very poor boiler can be made to show fairly good tests under expert manipulation.

A practical test of a boiler involves its testing under actual conditions of work through a very considerable space of time. The so-called laboratory tests of the past have been absolutely delusional as to the real economical merits of the appliance.

The great wastes in heating appliances result from imperfect combustion; and expert stoking of a very poor boiler will show a thermic efficiency in the boiler that can never be secured under the usual actual conditions of neglected intermittent firing and stoking. Under actual test the fuel used in all tests should be substantially of the same quality, and of the same size for tests of all boilers of the same size, and after the fuel charges are fed to the boiler, no freeing of the fuel charge from ashes should be permitted during the six or eight hours' run of the test.

Testing with the evaporative test is an objectionable method to secure accurate data. The test should in all cases be based upon the conveyance of heat units to feed water of a predetermined starting temperature, and measured by weight and bulk; the amount of water passing through each boiler of each specific size to be the same in all cases. The efficiency of the boiler part of the apparatus should be rated separately from that of the furnace part.

The astonishing fuel economies which can be effected through an equalization of the rate of combustion are enormous. This branch of the subject is as yet practically without cognizance on the part of the scientific world, but should be taken into account in estimating efficiency values of a practical nature.

The writer is strongly of the belief that with the propositions as to methods of rating as yet advanced, no satisfactory result is in sight, and he does not believe that any such satisfactory result can be secured except under general governmental control, involving an enormous expense for class benefit, which possibility is so remote as to be negligible.

There does not seem to be any way out of this quagmire, except through the individual commercial activity of the manufacturers. Just as soon as any manufacturer can demonstrate that he has a line of heating devices that are superior to the makes that are now sold, that manufacturer can afford to and will challenge his competitors to competitive tests under conditions that are indisputably fair. When that time arrives it should be the duty of a disinterested officially-appointed committee on testing to supervise such trial and make a fair report as to the result.

In the meantime, the heating engineering world ought to be able from the history of heating experience to state with fair accuracy how much grate surface is needed per square foot of radiation; how much direct heating surface is desirable per square foot of grate surface; how many square feet of indirect or flue surface is desirable, and what arrangement of flue surface is reasonably most effective. Such matter should be made public without direct reference to its applicability to any manufacturer's special designs. If any manufacturer happens to have designs in accord with this general statement, it is his business to advertise the fact to the public at large at his own expense. The main point I want to emphasize in this connection is: that under the proposed methods of rating boilers that so far have been elaborated by the non-manufacturing world, great injustice may be done to the designs of some makers, while undue endorsement may be given to unworthy competitors.

J. M. W. KITCHEN.

East Orange, N. J.

Origin of Vacuum Steam Heating

EDITOR HEATING AND VENTILATING MAGAZINE:

Could you inform me where I can obtain the best information in regard to the origin and development of vacuum systems of steam heating? I notice that several of these systems are described in Prof. Carpenter's work on "Heating and Ventilating buildings," in the "Principles of Heating," by Wm. G. Snow, and in a book on "Modern Steam Traps," by Stewart Gordon of London.

As a mater of historical record, I would like to know to whom should be given the credit of originating the first vacuum system. I am also very much interested in the development of these systems, and in a fair and impartial description of them as they are commercially sold to-day.

I noticed a few months since, that you had a statement emanating from the Paul Steam System Co., claiming that they still had some broad patents, and so if anything has been published in regard to the patent situation, will you kindly let me know where I can find it?

J. ADEE.

New York, Feb. 18, 1909.

Carrying Capacities of Steam Mains

EDITOR HEATING AND VENTILATING MAGAZINE:

The enclosed correspondence may interest your readers as a further contribution to the discussion of the carrying capacities of steam mains.

J. A. DONNELLY.

New York, March 2, 1909.

MR. JAMES A. DONNELLY,
132 Nassau Street, New York.

Dear Sir:—The paper written by you and published in THE HEATING AND VENTILATING MAGAZINE for January, 1907, has given me a good deal of pleasure, as well as a lot of information. I cannot get the same results as you do in some cases. For instance, the carrying capacity of steam mains according to Unwin is as follows:

1-in. pipe=40. You get 40.
1¼-in. pipe=75. You get 75.
1½-in. pipe=122. You get 126.
2-in. pipe=268. You get 286.
2½-in. pipe=497. You get 535.
3-in. pipe=820. You get 890.

I have worked these problems by arithmetic, and by logarithms and get the results as shown.

Another formula that bothers me is that for the velocities according to the various authorities. I do not understand how $D^3 \times 30$ can produce 12, 15, 18, 24, 30, etc., or how $D^3 \times 25$ produce 10, 12, 15, 20, 25, 30, etc. I have worked the velocities as follows: Take the carrying capacity of the different authorities from 1-in. to 16-in. and divide by the following factors:

For 1-in. pipe 100, 56, 45, 40, 30, 25, 30, $\div 2.5 =$ the various velocities. For 2-in. pipe 400, 317, 294, 286, 218, 200, 240, $\div 10.0 =$ the various velocities.

Regarding the size of steam mains for direct radiation, I figure that a 2-in. steam main will supply 200 sq. ft. of radiation 100 ft. away with a drop of .48 oz., if the velocity is 20 ft. per second. Or under the same conditions a 2-in. steam main will supply 1,000 sq. ft. of radiation if the velocity is 100 ft. per second.

I would like information and will ask you the following questions:

1. What is the correct number of sq. ft. of radiation from 1-in. to 3-in. as given by your article in THE HEATING AND VENTILATING MAGAZINE of January, 1907, using Unwin's formula? Is it what I have given, or what you give?

2. Give the formula, and work in full for finding the velocity of steam through a 2-in. pipe using Unwin's and D'Arcy's formulas.

3. What pressure on a steam boiler, delivering steam into the atmosphere indicates a velocity of 100 ft. per second?

4. On page 21 of this magazine for January, 1907, I notice the following:

0 lbs. pressure=90 ft. per second.
5 lbs. pressure=80 ft. per second.
10 lbs. pressure=72 ft. per second.
15 lbs. pressure=65 ft. per second.

Is it a correct statement that as you increase the pressure on a steam main, you reduce the velocity?

5. Under this same statement: What would be the velocity if this pressure was increased to 200 lbs. per sq. in.?

ALFRED D. HOLT.

Maynard, Mass.

MR. DONNELLY'S ANSWER

MR. ALFRED D. HOLT,
Maynard, Mass.

Dear Sir:—I am in receipt of yours of February 8, and am pleased to learn that the paper to which you refer has been of some use to you. It will give me pleasure to furnish you any in formation I may be able, upon this or any other subject that has to do with steam circulation, and I would therefore reply to your questions as follows:

The calculations in my paper were made as far as possible from the original formulas, and this is at all times necessary, if any degree of accuracy is desired. For instance, what is referred to as the Unwin formula, which you will find given by G. H. Babcock in "Steam," is as follows:

$$W = 87 \sqrt{\frac{W\,(p_1 - p_2)\,d^5}{L\,(1 + \frac{3.6}{d})}}$$

in which w is the weight of steam in pounds per minute, w, the density or weight, per cu. ft. of steam at the pressure p_1, which is the initial pressure; p_2, is the terminal pressure, the difference between them being the drop in pressure. d equals the diameter in inches; L equals the length in feet. If we change this formula, basing the calculation on steam at atmospheric pressure, with a drop in pressure of 1 oz. per 100 ft. run, and base the amount of steam carried as that necessary to supply direct radiating surface, having a condensation of 0.3 of a lb. per sq. ft. per hour, we obtain the formula:

$$R = D^{2.5} \times 87 \sqrt{\frac{D}{D + 3.6}}$$

as given in my paper.

It is purely an accident that the exponent 87 is the same in both cases. As a matter of fact, the exponent is not 87 when changed over to the second form, but it is so near it that the fraction may be practically disregarded. However, the Unwin formula which is given: $R=D^{2.75} \times 40$, is only substantially accurate.

You will notice in the text, that it is stated in the sizes from 1 to 16 in.; $D^{2.5} \times \sqrt{\frac{D}{D \times 3.6}}$ is substantially equal to $D^{2.75}$.

I think that you used this latter formula for your calculations, and of course it gives you a different result from the figures in the tables.

In regard to velocity of flow under the formulas: $D^3 \times 25$; and $D^3 \times 30$, etc., would say that these are not formulas of velocity, but formulas of capacity, and where they appear over the columns having to do with velocity, they must be regarded as only identifying the particular column. The velocity in each case was figured from the cubic feet carried and the area of the pipe, and no formula for obtaining these velocities is given in the paper. The foregoing, I think, answers your questions 1 and 2.

In regard to question 3, "what pressure on a steam boiler delivering steam into the atmosphere, indicates the velocity of 100 ft. per second," would say that your question is not completely the element lacking being whether the steam was to be delivered through an orifice in a thin plate, through a short tube, or through a pipe of considerable length. The basis of all the formulas of velocity is, as I think you are aware, $\sqrt{2gh}$.

In regard to question 5, which has to do with the first table given in the paper "Capacity of Steam Mains with Varying Pressures" would say that this table was figured, as you will notice in the text, on the assumption that the drop in pressure was to be the same; namely 1 oz. to 100 ft. where varying initial pressures were used, and the facts are correctly stated that as the steam is of higher pressure and more density, the velocity becomes less; the comparative velocities between fluids of different densities being proportional to the square root of the density of the fluids.

I trust that the foregoing has answered your questions, and think that if you will apply to the paper again that it will clear up the points about which you were in doubt. However, if there is any additional information which I can give you at any time, I shall be pleased to furnish it.

JAMES A. DONNELLY.

New York.

Typical Installation of Van Auken System of Vacuum Heating

Although the Van Auken system is commonly applied to heating systems used in connection with power plants, where exhaust steam from engines and pumps is the heating medium, it is equally designed for use in heating systems where no exhaust steam is available. An arrangement of the system under the latter conditions is shown in Fig 1.

Steam is distributed to the radiators, coils or other apparatus by a heating main and branches, as indicated. Each piece of apparatus is controlled by a valve on the steam supply. Corresponding to the steam main and branches is a vacuum return system leading to a vacuum pump near the boiler.

Belvac thermofiers (automatic relief valves) are placed on the return ends of all radiators or other heating apparatus connected with the system. Similar valves are placed at all points where condensation is likely to collect in the steam main. The vacuum pump is operated by steam from the boiler and maintains a constant vacuum of 6 in. or more throughout the vacuum return system, thus drawing all air and water of condensation from the units of radiation through the Belvac thermofiers, which prevent steam from passing into the return system and destroying the vacuum.

The speed of the vacuum pump is automatically governed by a pump governor, which is controlled by the vacuum in the return main.

The air and water are discharged into an automatic receiver and pump governor, from which the air escapes through a vapor pipe.

In this case the water is pumped into the boiler by a steam driven boiler feed pump controlled by a float valve on the receiver.

The exhaust steam from the pumps is separated from cylinder oil by an oil separator, and discharged into the heating system, where it is condensed and returned to the boiler. The oil is discharged to the sewer through oil trap.

As the amount of exhaust steam from the pumps is slight, the greater amount of the steam for heating purposes is reduced from boiler pressure to atmospheric pressure by a pressure regulating valve, which so regulates the amount of live steam admitted to the apparatus that exhaust steam is utilized to its fullest extent before live steam is admitted.

When special conditions make the use of low pressure boilers advisable the apparatus can be proportioned to operate with a boiler pressure at 10 lbs. or less.

Fig. 2 shows the application of the Van Auken system to a power-plant where an open feed water heater is used.

The feature of the Van Auken system is, of course, the automatic relief valve, which is called the Belvac Thermofier. This valve is connected to radiators or coils at the point K (shown in the sectional illustration of the valve). The return pipe to the vacuum pump is connected at B^2.

The water of condensation is drawn through passage D^2 and accumulates in pocket L until it reaches the inverted weir E, it is then drawn upward in space D until it overflows into the float chamber A A, here it accumulates until the line of floatation is reached. This causes the float C to lift, opening the valve seat at H H which allows the water to escape into the vacuum return pipe B B. After

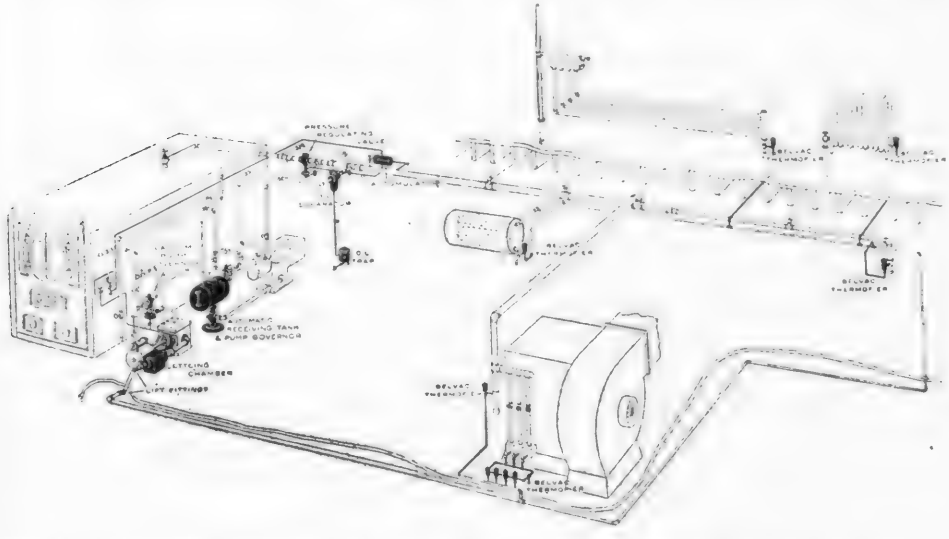


FIG. 1.—A COMMON METHOD OF INSTALLING THE VAN AUKEN SYSTEM IN HOTELS, OFFICE BUILDINGS, ETC., WHERE NO ENGINES ARE USED

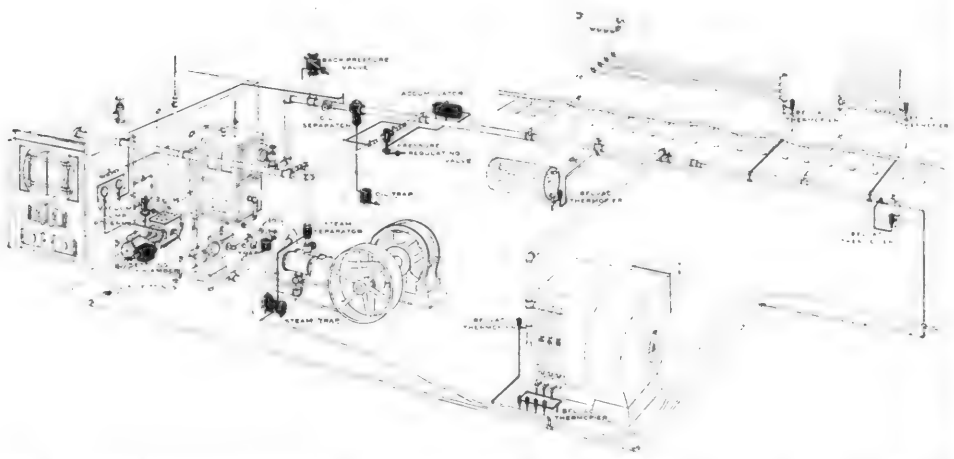


FIG. 2.—APPLICATION OF VAN AUKEN SYSTEM TO A POWER PLANT CONTAINING AN OPEN FEED WATER HEATER

the remomal of the water the float again settles on seat H H until sufficient water accumulates in float chamber A A to again lift the float.

The air contained in the radiators or coils, due to condensation, enters through the column of water in space D into the top of float chamber A A. Here its direction follows arrows G G, being drawn through the small opening in guide-pin at F down through the hollow body of copper float, past the valve seat at H H into the vacuum return B B^2. This removal of air is continuous regardless of the amount of water present.

The by-pass I when opened allows all dirt, core-sand or scale to pass directly into the vacuum return, thus cleaning the valve and preventing clogging by any foreign substance; the arrangement of by-pass is such that the water may be emptied from chamber A A without interfering with conditions existing in space D.

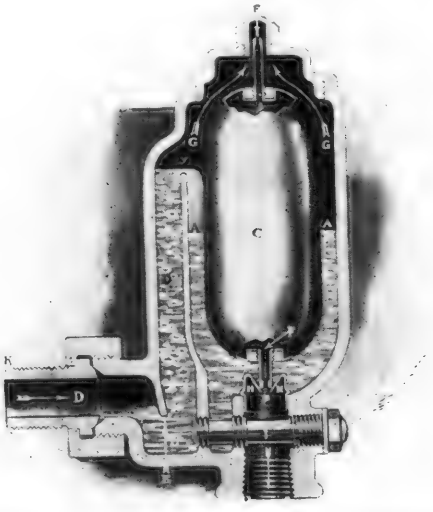
There is no possibility of steam escaping into the vacuum return as the column of water at space D acts as a water seal, which even in the smallest thermofiers, is 3 in. deep; also the opening on top of guide at F is so constructed that steam cannot enter. The pin upon which this travels insures an opening free from foreign matter at all times.

The upper and lower guide valve pin are interchangeable, making it immaterial as to how the float is placed in the valve as it works successfully either way. The pin or guide projecting from the cap on top of the valve, keeping the float in proper position. The air outlet being free at all times and the accumulation of foreign matter in the air passage is therefore impossible.

The construction of the Belvac thermofier is simple, its automatic action relying entirely on the amount of water being present, due to the condensation of the steam. There is no reliance on springs, pistons or sliding fits of any kind, nor any dependence on expansion or contraction of rubber or composition for its operation.

The float chamber is accessible by removal of valve bonnet which can be done at any time during the operation of the system.

The by-pass is so constructed that one can observe by the outward appearance as to whether the by-pass has been tampered with, as the by-pass head should always be tight against the valve body in order that that the thermofier may give the best results. The Van Auken system is controlled by the Consolidated Engineering Co., Chicago.



BELVAC THERMOFIER, OR AUTOMATIC RELIEF VALVE, USED WITH VAN AUKEN SYSTEM OF VACUUM HEATING

TRADE AND MISCELLANEOUS NOTES

Deaths

Thomas J. Waters, chief engineer of the Board of Education, Chicago, since 1884, died in that city February 25, after a brief illness, of pneumonia. He was 59 years old. Mr. Waters was a charter member of the American Society of Heating and Ventilating Engineers and was widely known in the heating trade throughout the country. He had been connected with the Chicago Board of Education since 1883, commencing as one of the board's engineers. He leaves a widow, one daughter and three sons.

Mrs. Thomas N. Thomson, wife of Thomas N. Thomson, of Scranton, Pa., died February 17, in her 41st year.

Miscellaneous Notes

Richmond, Va.—The Richmond Academy of Medicine has adopted the following resolutions, calling attention to the danger of gas in sleeping rooms:

"Whereas, it is a well known fact that the products of the combustion of fuel of any kind are poisonous to the human system if breathed for any length of time; and

"Whereas, in the case of gas and oil stoves, used for heating purposes, there is seldom any provision made for the escape of these products except into the room; and

"Whereas, the public in general seems to be ignorant of these facts; therefore be it

"Resolved, That the Richmond Academy of Medicine and Surgery regards the use of gas and oil stoves for the heating of rooms in which persons remain for any considerable length of time as decidedly prejudicial to health, unless provision is made for the escape of the gases of combustion in the same manner as in the case of stoves which burn coal or wood.

"That the members of the academy use their influence, both individually and collectively, towards acquainting the public with these facts and, especially, towards warning their patients against the use of gas, and oil stoves for the heating of bedrooms and living rooms."

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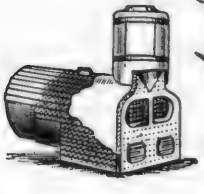

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Cleveland, O.—Changes and additions to the plans for heating and ventilating the new court house will increase the cost of the plant $56,000. The total expenditure for the work will be $92.000. Andrew Dall & Son, Cleveland, have been awarded the contract.

Frank W. Mills, formerly associated with Abendroth Bros., Portchester, N. Y., has become identified with the Globe Foundry Co., Portchester, of which his brother, W. A. Mills, is a member.

O. P. Simler, secretary and general manager and one of the organizers of the Union Radiator Co., Johnstown, Pa., has resigned. Mr. Simler will continue to be identified with the radiator business.

Building operations for January showed a remarkable increase over those for the same period last year. In but one city out of 35 was there a decrease, while the increases ran from 120% to 140%. The Eastern cities showed an increase of 120%; middle west, 120%; and south 120%. New York City, including all boroughs, reported a total $14,839,609 as against $6,850,855 last year.

Little Rock, Ark.—Central Heating & Mfg. Co., Little Rock, will add one 500 H.P. boiler and one 5,000,000 gal. centrifugal pump to its equipment by September 1 next. The plant was released from the hands of a receiver last December after the debts of the concern had been adjusted.

American Heating and Ventliating Co., Eau Claire, Wis., has filed notice of dissolution.

Richmond, Ind.—Plans are being perfected by local capitalists to establish a central hot water heating system in Richmond at a cost of $500,000. It is planned to commence work in the spring.

Minneapolis, Minn.—An ordinance has been introduced in the city council giving a franchise to operate in Minneeapolis to the Northern Heating and Lighting Co. About a year ago the same company made application for a franchise, but the ordinance, after being referred to committee, was not heard of again.

Dayton, O.—Stockholders of the Springfield Light, Heat & Power Co., Dayton, have decided to issue bonds in the sum of about $650,000 to carry out in detail the merger resulting from the taking over of the Kelly plant.

Henry Aird, Troy, N. Y., former president of the Eastern Supply Association, was presented by the Association with a grandfather's clock as a testimonial of the association's regard for him. The gift was sent to Mr. Aird's home and was formally acknowledged by him in an address delivered at the recent meeting of the association in New York. Mr. Aird,

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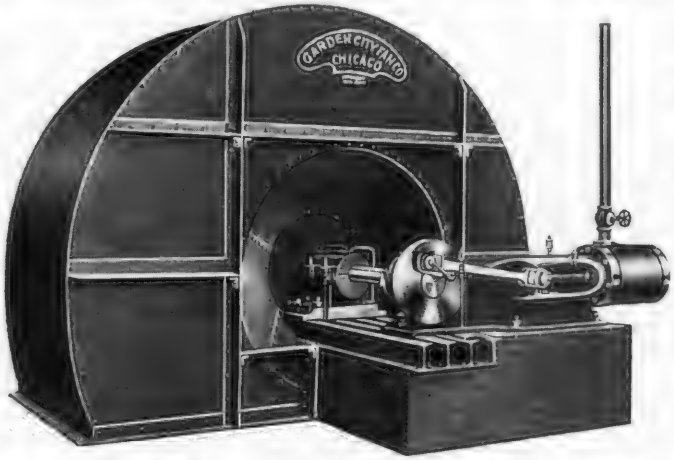

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who has been in poor health for some time, left for Pinehurst, N. C., after the meeting.

Federal Furnace League has issued its handbook on furnace heating. It bears the title, "The Warm Air Furnace." The publication of the handbook is the outcome of the efforts made by the league to uplift the art of furnace heating and the volume contains much matter in the shape of practical rules, methods and suggestions on furnace installation. The book is issued from the office of the secretary of the Federal Furnace League, Wilson Ferguson, Mint Arcade, Philadelphia, from whom copies may be procured without charge by furnace dealers, architects, contractors and others interested.

Federal Furnace League will hold its annual meeting at Atlantic City, N. J., in May next, when a number of papers on the furnace industry will be presented.

Portland, Me.—The annual dinner of the Plumbing, Steam and Metals Salesmen's Association was held at the Congress Square Hotel, Portland, March 8. A vaudveille programme added to the enjoyment of the occasion.

Manufacturers' Notes

Abendroth Bros., Portchester, N. Y., has moved its executive offices to its New York City quarters at Pearl and Beekman streets.

John D. Hibbard and George F. Hughson, vice-president and secretary, respectively, of the John Davis Co., Chicago, Ill., announce their withdrawal from the company. Mr. Hughson has organized the Hughson Steam Specialty Co., which will make the steam traps and specialties heretofore made by the John Davis Co. He will also represent the A. M. Byers Co., of Pittsburg, pipe manufacturers. Mr. Hibbard has not announced his future plans.

H. W. Johns-Manville Co., suffered a loss of $200,000 February 13, when its branch warehouse in Milwaukee, Wis., was destroyed by fire. Six persons were killed and a number were injured. Plans have already been drawn for a new factory and office building in Milwaukee, to cost $150,000. It will be 100 x 150 ft., and six stories high. The company has secured temporary quarters in the Menomonee Public Service building.

American Radiator Co., Chicago, will build an addition to its Pierce plant in Buffalo, to be occupied as a storehouse. The addition will be 100 x 115 ft., one-story high, of steel and corrugated iron construction.

American Radiator Co., Chicago, announces an increase in dividends, the

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quarterly dividend on the common stock being advanced from 1% to 1½%. This action places the company's common stock on a 6% basis. The usual extra dividend on the common stock was also declared, making the total yearly rate 8%. A quarterly dividend of 1¾% was declared on the preferred stock.

G. M. Davis Regulator Co., Chicago, reports a recent shipment to the General Fire Extinguisher Co., Providence, R. I., of a 30-in. pressure reducing valve to reduce pressure of 75 lbs. down to 30 lbs. It is designed to pass 20,000,000 gals. of water per day. The shipment weighed 3 tons and the valve is considered to be the largest pressure reducing valve ever constructed in this country. Further details of this big regulator are promised for a later issue after the regulator is placed in operation. The company also reports an order for a 30-in. combination atmospheric relief and back pressure valve to be used on a 5,000 K.W. Curtis turbine installed in the 59th street station of the Interborough Rapid Transit Co.'s plant in New York City. This company has 19 30-in. Davis relief valves installed in its plant which is considered the largest power plant in the world and containing the largest installation of one type of relief valve.

This valve is designed to be tight against 29 in. of vacuum and to relieve at atmosphere or any predetermined pressure above atmosphere for which the valve may be set. The change from vacuum to pressure conditions in operation is entirely automatic and is the feature that have made Davis combination valves the standard for outboard exhaust lines in connection with turbine units.

New Incorporations

Dakota Plumbing & Heating Co., Grand Forks, N. D. Capital stock, $10,000. Incorporators, Hans C. Nelson, Peter Braseth and Andrew Peterson, all of Grand Forks, N. D.

W. H. Smith Co., New York, capital $10,000, steam heating and ventilating contractors. Incorporators: James Gallani, William H. Downs, J. J. Kramer, 220 Broadway.

A. J. Anderson & Co., Louisville, Ky., capital $60,000, to manufacture heating, lighting and refrigerating apparatus and pneumatic and hydraulic machinery. Incorporators: A. J. Anderson, Cincinnati; Walter D. Krauth, New Albany; C. P. Lichty, Louisville.

Graves Plumbing & Heating Co., Wilmington, Del. Capital stock, $20,000. Incorporators: William M. Graves, Ethel M. Graves and Martin I. Traynor, all of Wilmington.

American Heating and Ventilating Co., Richmond, Va., capital $10,000, to conduct a heating, ventilating and plumbing business. President, C. A. Haslett, Philadelphia; vice-president, W. R. Murphy, Richmond; secretary and treasurer, L. G. Whitmer, Bloomington, Ill.

Akron Automatic Hot Water Heating Co., Akron, O., capital $10,000, to manufacture hot water heaters. Incorporators: Charles A. Levy, James W. Meeker, W. W. McIntosh, Lester A. Teits and J. L. Reid.

Central Heating System, Newark, N. J., capital $10,000, to furnish steam power for machinery, heating, etc. Incorpora-

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tors: Edward S. Prieth, Benedict Prieth and Charles E. C. Smith.

Standard Ventilating Co., New York, capital $15,000, to manufacture hardware and building fixtures for ventilating purposes. Incorporators: Isaac L. Adleman, 150 Nassau street, New York; Clifford N. Worth, Yonkers, N. Y.; and Ulysses D. Everts, 111 Chestnut street, Brooklyn, N. Y.

Eastman Heating Co., Boston, Mass., capital $5,000, to conduct a heating and ventilating business. President, Charles H. Eastman, 406 Atlantic avenue; treasurer, Charles W. Sisson, 135 Atlantic avenue, Boston.

Campbell Heating Co., Des Moines, Ia. Capital stock, $30,000. Incorporators: H. D. Campbell and A. S. Campbell.

Hydrostatic Machinery Co., Rochester, N. Y., capital $50,000, to manufacture steam fitting and plumbing supplies. Incorporators: Louis J. Ernst, Adolph M. Spiehler and Clarence Wheeler, all of Rochester.

Pilot Heater Co., Lorain, O., capital $100,000, to manufacture heating supplies. H. William Pilot is the principal.

Chicago Contracting & Engineering Co., Chicago, Ill., capital $30,000, to install and operate power and heating plants and operate machine and repair shops. Incorporators: John M. Blakely, Arthur B. Wells and H. H. Talcott.

Sheridan Plumbing & Heating Co., Sheridan, Wyo., capital $10,000, to conduct a heating and plumbing business. President, W. T. Young; vice-president, Charles Dow; secretary-treasurer, L. B. Glafke.

Sanitary Heating Co., Brooklyn, N. Y. Capital stock, $400,000. Incorporators: W. H. Scofield, W. J. Wilson, F. L. Hulse, F. L. Shiffey and W. E. and F. R. Wilson.

Eckhart Plumbing & Heating Co., Seattle, Wash., to take over heating and plumbing business of the Ernst Hardware and Plumbing Co. The new firm is composed of William F. Eckhart, Claude H. Eckhart, Seattle, and Gilbert Zindle, Akron, O.

American Thermostat Co., Elmira, N. Y., capital $25,000, to manufacture the Marvel Thermostat. It is proposed to build a plant with a capacity for manufacturing 100 themostats a week. Incorporators: Burchard C. Johnson, Arthur W. McClellan and Byron C. Johnson.

Business Changes

Washington, D. C.—Sealed proposals will be received at the office of the Supervising Architect, Treasury Department, for the following-named work:

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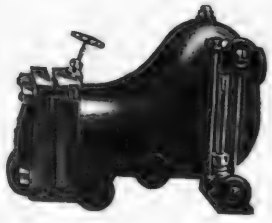

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Contracts Awarded

A. J. Rogers Co., Paterson, N. J., steam heating, plumbing and sheet metal work in the new Lakeview School No. 22 in Paterson, for $6,000; Public School No. 22, for $11,000; School No. 23, for $15,000, and the Home for Incurables at Parmies, N. J., for $13,000.

Olean Plumbing Heating & Lighting Co., Olean, N. Y., heating, plumbing and lighting systems in the State School of Agriculture at Alfred, N. Y., for $0,000; also heating, plumbing and lighting system in the new $50,000 clubhouse of the Knights of Columbus in Olean.

Gillis & Geoghegan, New York, power, heating and ventilating plant for the Emma Willard Seminary buildings in Troy, N. Y. The contract amounts to $52,600. A forced hot water system will be installed according to plans drawn by Henry C. Meyer, Jr., New York.

Willey & Calhoun, Portland, Me., heating barracks and non-commissioned officers' quarters at Fort McKinley for $5,218; A. H. Moulton, Portland, Me., has the heating contract, at $1,039, for the bachelor officers' quarters.

E. Fay & Son, Philadelphia, heating new public school at 26th street and Cumberland streets for $5,000.

McGinness-Smith Co., Pittsburg, Pa., heating 12-story building on Liberty avenue, Pittsburg, for Herbert L. May. A 9,600 sq. ft. Mercer boiler will be installed.

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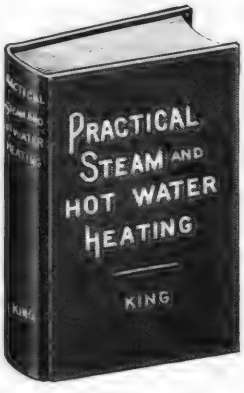

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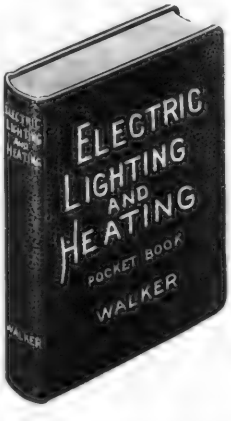

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Gallaher & Speck, Chicago, Ill., steam heating and power plant for the new county infirmary which will be built by by the Alling Construction Co. The power and heating contract amounts to $11,000.

Henshaw, Bulkley & Co., San Francisco, Cal., heating and ventilating Garfield Primary School for $11,000.

Pettit-Galloway Co., Little Rock, Ark., hot water heating plant for the new Ada Thompson Memorial Home, for $2,100.

Dewstoe-Brainerd Co., Cleveland, O., ventilating system for the post-office building.

Beasley & Miller, Pittsburg, Kan., heating and plumbing new Y. M. C. A. building at that place for $5,000.

Frederick A. Tyler, Albany, N. Y., steam heating coaling station and machine room at Suspension Bridge for the N. Y. C. & H. R. R. R. Mr. Taylor has just completed a similar contract at Rennsalear. Also heating 23 houses on the Manning Boulevard for U. C. Stockwell, and supplying 10 heaters for buildings on South Pine avenue being built by Lorenz Willig.

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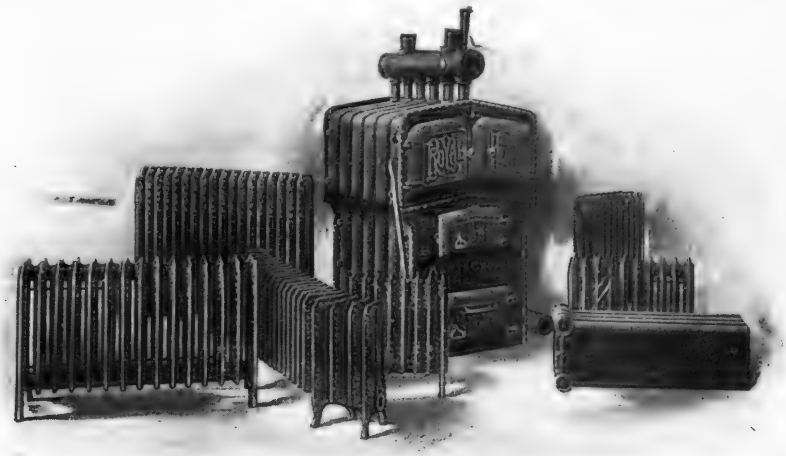

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